# PRODUCT ENGINEERING

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## *New Deal, New Products*

DOES the New Deal mean the end of industrial initiative and ingenuity; a moratorium on engineering and product development? Emphatically no, according to the National Recovery Administration, leading interpreters of the New Deal. In a recent speech before the Advertising Club of New York, Deputy Administrator Malcolm Muir had this to say:

"I see in this new scheme of things a challenge to our inventive genius, a greater opportunity than ever before for the development of new products, to be manufactured by new low-cost processes, to be bought by the great new masses of purchasing power.

"I want to emphasize the opportunities the future should hold for merchandising and advertising ability. Whether we like it or not, we are faced with a realignment of purchasing power. We must find out where this purchasing power will be, and what it will buy. We must chart a new consumer map. . . ."

The search for new markets that Mr. Muir pictures so interesting to advertisers and merchandisers will be even more fascinating to the engineer. His will be the job of anticipating the desires of the new masses of purchasing power, because he must design the products for the advertisers to shout about and the merchandisers to sell.

Because of the sweeping changes that are taking place it is impossible to forecast the nature of the changes that product designs will undergo. But again quoting Mr. Muir, ". . . You and I well know that the price cutter was the biggest builder of profitless prosperity. . . ." As is known only too well by most engineers, the price cutter was also the foe of engineering quality in the product. His elimination marks the dawn of a new day in product design, and it is safe to forecast that in the immediate future buyers will place more emphasis on inherent design quality and less on penny-pinching shoddyness. Many designers now hemmed in by price restrictions will welcome the change.

## *Don't Hang Around Your Own Back Yard*

IT DOES not take long for the young graduate engineer to discover that his future progress is likely to depend, in large measure, on his ability to continue to broaden his professional knowledge. And to continue to learn he must find sources of information and inspiration. He can learn much from the seniors of his own profession. *Product Engineering* and similar engineering periodicals were established to make available other sources. Engineering society meetings are likewise helpful.

But thoughtful engineers do not stop at domestic sources. They also study the significant developments in engineering technique that come from abroad. This is indicated by the surprisingly large demand for the engineering society monographs sponsored by the four leading engineering societies and published by the McGraw-Hill Book Company. And the authors of these highly technical monographs are Nadai, Tietjens, Timoshenko and Bakhmeteff, all of them men of foreign birth and foreign professional training. Every meeting of the Applied Mechanics Division of the A.S.M.E. features papers by European trained engineers.

With all due respect to American institutions and traditions, engineers must not permit themselves to be hemmed in by a false sense of patriotism or egotistical nationalism. For him who seeks the truth, the whole world is a hunting ground.

# Ball Bearing Mountings for High-Speed Spindles—I

An outline of the factors to be considered in the design of ball bearing mountings for spindles running between 8,000 and 50,000 r.p.m., illustrated with nine typical designs. Eleven additional designs will be presented in the October number

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IN THE design of ball bearing mountings for spindles rotating at extremely high speeds, such as in the range from 8,000 to 50,000 r.p.m. or more, many factors that can be ignored safely in slow-speed spindle mountings must be given special consideration. Not only must the bearings function properly when first assembled, but what is of greater importance, the life of the bearings must be insured against early failure. All of the effects of centrifugal forces, vibration and wear are greatly accentuated by the high speed. Bearing selection, mounting accuracy, bearing adjustment for the take-up of axial play, sealing against foreign material and the problem of lubrication are the cardinal points to be given consideration.

The accompanying drawings show the ball-bearing mountings for internal grinder wheel spindles which operate in the high-speed range. The designs illustrate how the requirements have been met successfully by a number of engineers. As the same fundamental conditions prevail for all types of spindles, the designs shown here have possible applications to any form of shafting that must revolve in the same speed range.

Numerous different types of ball bearings have been used for the mountings shown in these designs. But no attempt shall be made here to advise the selection of one particular type of bearing in preference to another. Catalogs and handbooks of the bearing manufacturers will help in making a preliminary selection. But in every instance the designer should seek the advice of the manufacturer's engineering department and follow rigorously the recommendations given.

There are several general features that are usually incorporated in every design of bearing mounting for high speeds. For such service precision bearings, in which the S.A.E. tolerances have been halved, are usually recommended. Precision bearings are not only built to a superior degree of accuracy, but they are also usually assembled selectively in order to practically eliminate all play. They cost slightly more than standard bearings. Another factor of great importance in the design of high-speed spindles is the rigidity of the spindle parts. This is obtained by using generous dimensions in both rotating and non-rotating pieces, the greater masses tending to absorb and dissipate vibrations.

Accuracy in machining the various details must be stressed. Bearing seats and shaft shoulders must be ground; also the clamping nuts and spacers, especially on the end faces. It is of particular importance to follow closely the bearing maker's recommendations relative to bearing fits in the housing bore and on the shaft diameter. If the fits are too tight, the outer ring of the ball bearing is compressed and the inner ring is stretched. This will have a tendency to clamp the balls between the two races. If, on the other hand, the fits are too loose the inner ring of the bearing will rotate on the shaft and the outer race will not have the proper support in the housing. Both of these conditions are not permissible. Loose bearing fits usually cause whipping of the shaft, pounding of the bearings, noise or misalignment.

Balance is another important consideration. For this reason, keys, keyways, spanner holes and the like should be in multiples and equally spaced circumferentially at the same radius distance from the center of the shaft. Screws should fill the holes that accommodate them so that the top of the screw will be flush.

Sealing the bearings to guard against the entrance of grit, water and other foreign material detrimental to bearing life is always important, but particularly so in high-speed bearings. It is reliably stated that fully 75 per cent of ball bearing failures are the result of foreign matter getting into the bearings either during the assembly of the machine or while it is in service. In the typical design given here various types of seals which have been used successfully are shown.

With reference to lubrication, regardless of the speed at which the ball bearing rotates, oil consumption is a negligible factor. But the quality of the lubricant must

be of the highest standard obtainable, neutral both to acid and alkali, both of which will attack and corrode the mirror-like surfaces of the balls and races. Mineral oil is used in preference to vegetable oil, as the latter becomes rancid. Sperm oil is an excellent lubricant and is used to a great extent. Concerning the method of lubrication, according to one engineer, when oil is dropped on a shaft revolving at 8,000 r.p.m. or more, centrifugal force throws off the drops in large globules which cannot properly enter the bearing. He therefore recommends and uses wick feed lubrication. But it is the experience of another engineer that at extreme speeds such as more than 50,000 r.p.m., the oil dropping on the rotating spindle is atomized, causing a fog of lubricant which gives to the bearing a limited but substantial amount of lubricant. Regardless of any conflicting ideas on the subject, oil that is passed through felt or wicking is effectively strained of all gritty impurities, which is an extremely valuable feature.

One of the first factors to be considered in the design of bearing mountings is alignment. Two general types of misalignment are possible. As in three bearing mounting, the centers of the three bearings may not lie in the same straight line. The other type of misalignment is where the plane of the bearing is not perpendicular to the axis of the shaft. In both types the misalignment might be caused by inaccurate machining. But in addition to this the deflection of a spindle running at high speed might cause a misalignment with reference to the perpendicularity of the plane of the bearing relative to the deflected axis of the rotating spindle. Such a condition has no relation to the accuracy of the machining and can only be taken care of by provisions inherent in the design. Self-aligning bearings solve this problem in a simple manner.

In all high-speed spindle bearings, it is essential that there be continuous metal to metal contact between the balls and the races. This can be accomplished by one of three general methods. An automatic adjustment feature can be provided by some form of spring backed floating sleeve or collar. The collar pressing axially against the floating outer race keeps the balls in contact with both races. Another method is to rely on periodic manual adjustment to take up the play. The third method is to use preloaded ball bearings.

The main advantage of the spring adjusted ball bearings is that they need no further manual adjustment during the life of the bearing. This type of adjustment, however, has its limitations. If the springs do not create sufficient pressure, the balls leave their angular contact with the races and drop down into the bottom of the groove in the races, causing a chance for vibration. On grinding spindle mountings this would cause a short speed hesitation, resulting in chatter marks on the work. Also, spring adjustment cannot be used for heavy end thrust and large radial shock loads.

Preloaded ball bearings have the advantage that no adjustment, either automatic or manual, is required during the life of the bearing. In the manufacture of preloaded ball bearings, a definite axial load is placed against one of the races during the operation when the faces of the bearing races are being ground. The loading compels the balls to make an angular metal to metal contact with the races. The faces of the bearing are then ground flush with one another. When the bearings are assembled into the machine and tightened into place, the "preload" will be duplicated automatically, the bearings being of course used in pairs.

In Fig. 1 is shown a typical construction using self-aligning bearings throughout. As the name implies, the self-aligning bearings find their own centers but they are not capable of supporting any large amount of end thrust. This bearing mounting is used in the Taylor & Fenn bore grinder.

Bearings *D* and *E* are self-adjusting after being assembled. This is accomplished through the spring backed slidable sleeve *F*, the springs *S*, of which there are six, pushing against the rigidly held collar *R*. In order that the springs will not tend to buckle, guide pins are placed inside them. These guides are simply nails of the proper diameter clipped off at the pointed end to the desired length, the nail head serving as a spring seat.

Shoulder collars *B* and *B* facilitate the removal of the bearing for repair by offering a shoulder to press against instead of applying a harmful pressure against the outer races.

Lubricant for all bearings is introduced through the screened felt at *M*. Labyrinth seals are provided at each end, the clearance between the seal units not exceeding 0.010 in. In this construction the belt pull is taken almost entirely on one bearing.

Ordinarily the use of three ball bearings, spaced an appreciable distance apart on the same shaft, is not considered the best policy because of machining difficulties in attempting to obtain accurate alignment. But in this construction much of the difficulty was circumvented by grinding the bore of the housing *A* to the same diameter throughout. This eliminated to a great extent the possibility of misalignment as in a shoulder bore at each end of the housing. This particular spindle has given excellent service over a long period of time.

Where considerable end thrust is encountered, as well as radial load, deep-groove ball bearings must be pro-

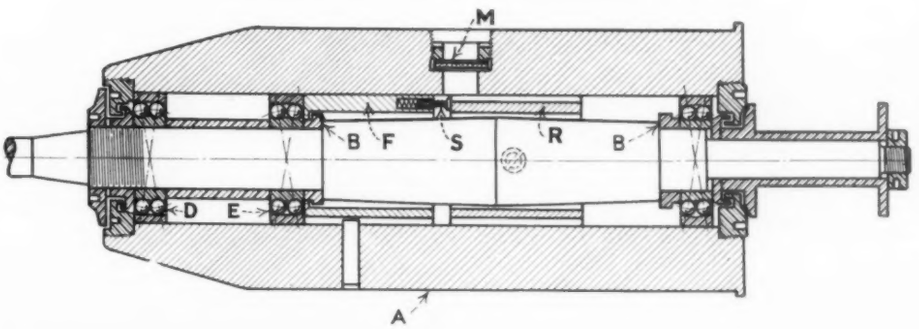



**Fig. 1 — A typical construction using three self-aligning bearings. Machining difficulties were largely circumvented by grinding the bore of the housing to the same diameter throughout, thus assuring axial alignment of the bearings**

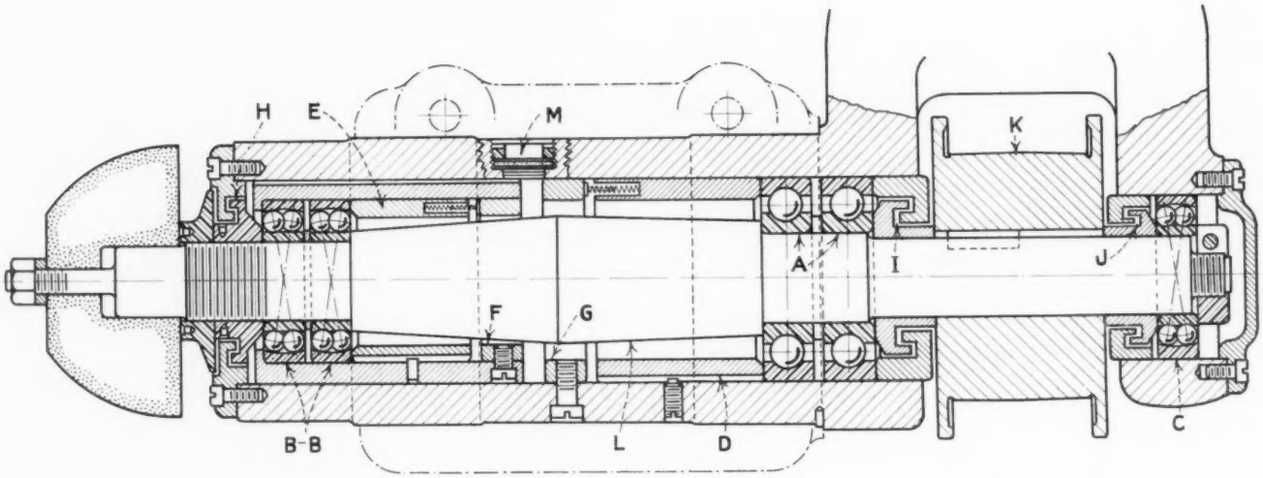



Fig. 2—End bearings are of the self-aligning type. The spring-backed sleeves make the bearings self-adjusting to take up end play

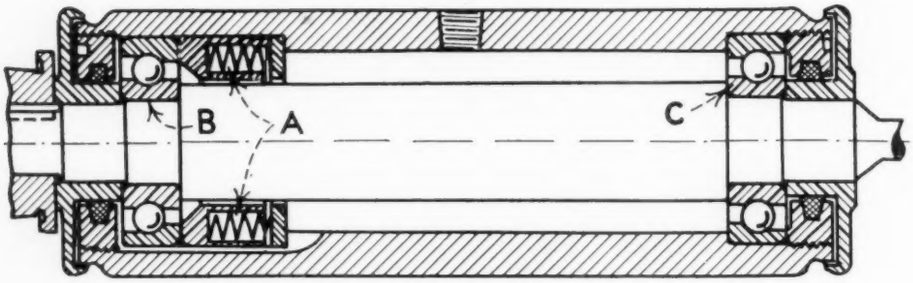



Fig. 3—A two-bearing mounting wherein a spring-backed collar takes up all the possible axial play in the ball bearings

vided to take the axial thrust. In Fig. 2 is shown the bearing mounting used by Taylor & Fenn in their bore grinder. In this construction two self-aligning bearings are employed together with a pair of deep-groove ball bearings to take the thrust.

As in the construction shown in Fig. 1, in this design the bearings *A-A* and *B-B* are self-adjusting after being assembled. The slidable sleeves *D* and *E* are pushed against the outer races of bearings *A* and *B*, respectively, the collars *F* and *G* being held rigidly.

Lubricant is introduced through the screened felt at *M* and labyrinth or tongue-and-groove seals are used, being denoted by *H*, *I* and *J*. The driving pulley *K* is keyed to the spindle, three keyways being cut in the pulley bore for balance. The outboard bearing in the overhanging arm gives additional support against belt pull and causes less deflection of the spindle. The spindle *L* is crowned to prevent whipping.

In Figs. 3 and 4 are shown two more spindle mounting designs wherein helical compression springs are used to obtain self-adjustment of the bearings. These spindles were made and successfully operated for a number of years by the Gurney plant of the Marlin-Rockwell Corporation, and were used on grinders for grinding the bores of precision bearings.

In the design shown in Fig. 3, two ball bearings are used. Both are spring adjusted through the series of springs *A*. They cause the collar in which they are mounted to press against the outer race of the rear bearing *B*. This axial push is transmitted through the bearing to the spindle, pulling the whole spindle rearward until the inner race of the front bearing *C* forces the balls against the outer race, thus pressing the latter against the shoulder in the housing. The initial com-

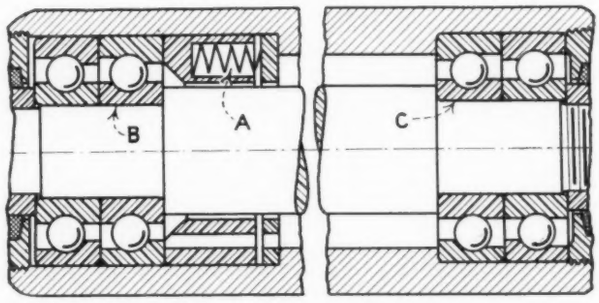



Fig. 4—This spindle construction is identical with that shown in Fig. 3 with the exception that two bearings are used at both the front and rear end, making it adaptable to higher speeds

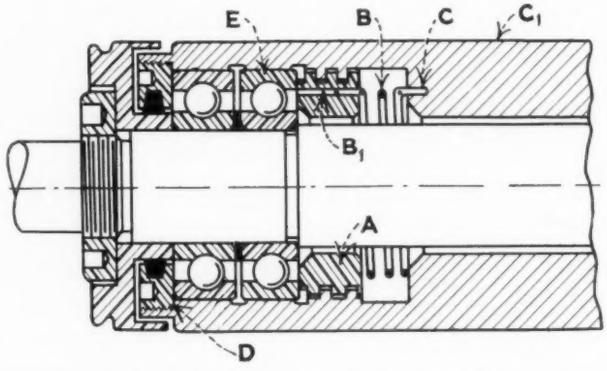



Fig. 5—A design of automatic bearing adjustment wherein a torsional spring anchored in the housing exerts a twist on a loosely fitted threaded collar which pushes against the outer race of the bearing

pression of the springs is such that they exert a total axial force of 100 lb., this being sufficient for relatively light thrust loads.

Formerly, this spindle was made with spirally grooved collars in place of the felt seals shown in Fig. 3. The purpose of the spiral grooves is to screw out dirt and grinding chips that were bound to try to enter.

In this particular design, felt was substituted for the spiral groove and it was found that the life of the bearing was increased approximately four times. In addition to this, the incorporation of a non-metallic ball retainer cage also contributed to the increased life.

In Fig. 4, the spindle construction is identical with that shown in Fig. 3, with the exception that two bearings are used at both the front and rear. As the pressure of the adjusting spring should be as large as possible, but within the range of the bearing capacity, the use of the double bearing has the advantage of permitting a larger spring load. Another advantage of this double mounting is that it is better adaptable to higher speeds as smaller bearings can be used, which consequently reduces the surface speed at which the balls must run. This type of design can be used for speeds up to 13,000 r.p.m. largely because of the smaller bearing size made possible by the use of the double mounting.

In some machines it is particularly desirable that the automatic bearing adjustment be designed so that it will be possible to change the axial pressure on the ball races whenever desired. A design of automatic bearing adjustment that accomplishes this effectively is shown in Fig. 5. It is used in the wheel spindle mounting of one of the grinders of the Bryant Chucking Grinder Company.

Whereas the designs previously illustrated made use of helical compression springs, in this design a loosely fitted threaded collar *A* is subjected to the twisting moment of a torsional spring *B* which is anchored in the housing at *C*. After the nut *D* has been screwed into place, the spring is released causing the collar *A* to unscrew until it bears against the outer race of bearing *E*. With a number of holes in collar *A* or housing $C_1$, the spring tension can be adjusted to the desired axial pressure.

In Figs. 6 and 7 are shown designs wherein preloaded ball bearings are used. Such bearings must always be used in pairs. When spacing collars, such as *A* and *B* shown in Fig. 6, are used, they must be of identical width and must be specified to be ground in pairs in order to assure exactly equal width. Otherwise the amount of axial thrust will not be the same as the preloading to which the bearings had been ground.

In the design shown in Fig. 6, the races of bearings *C* and *D* are clamped rigidly while bearings *E* and *F* float. For lubrication, oil is poured into the reservoir *G* to a level that will bring it to the center of the lower ball. The oil finds its way to the bearings through the holes *H* and *I* which are plugged with felt wicking to strain the oil. If an oversupply of oil is put in the reservoir, the excess is drained out through holes *J* and *K* which are also plugged with felt wicking in order to keep out dirt. The oil reservoir can be emptied by taking out the pipe plug in the bottom.

Labyrinth seals are at each end to prevent the entrance of dirt and the escape of oil. The pulley is attached to the tapered end of the spindle without the use of a key.

Preloaded ball bearings are also used in the construction shown in Fig. 7. One of the features of this design is the efficiency of the labyrinth seal. The slingers *A* and *A* rotate with the spindle and by centrifugal force throw off all grit or foreign matter that might attempt

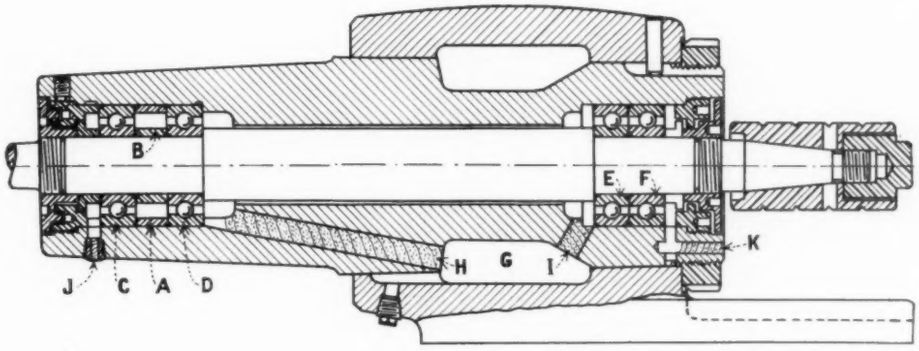



Fig. 6—Design of a spindle mounting wherein preloaded ball bearings are used, the bearings being mounted in pairs. Care must be taken in grinding the spacing collars *B* and *A*, in order to make sure that they will be of exactly equal width

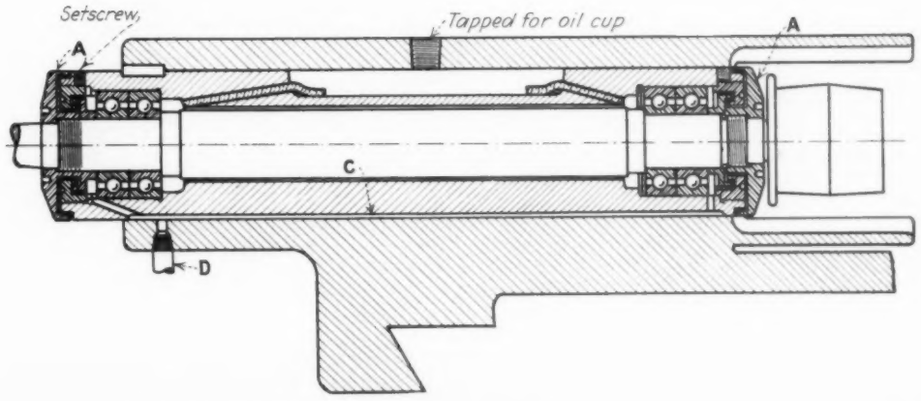



Fig. 7—Another design using preloaded ball bearings. The feature of this design is the efficiency of the labyrinth seal and the location of the oil reservoir which is placed above the bearing and is supplied through a sight-feed oiler

to find its way through the seal. Another feature of the design is the location of the oil reservoir which is placed above the bearing and is supplied with oil through a sight feed oiler attached at the point designated. Oil from the reservoir feeds by gravity through the wicking and collects in the groove *C* from whence it flows out at *D*. This overflow being out through the pipe, there is no opportunity for grit to enter at this point.

When the distance between two bearings is relatively great and wide variations in the temperature of the spindle are to be expected, some provision must be made for expansion and contraction. An example of a mounting design wherein such a condition is taken care of is illustrated in Fig. 8 which shows the spindle of the Dumore No. 5 grinder. This spindle is designed for speeds up to 40,000 r.p.m. At this high speed the spindle must be in as nearly perfect dynamic balance as possible. For this reason, in the final grinding operation the allowable eccentricity of the various diameters is limited to a maximum of 0.00025 in. The spindle is made of a tough heat-treated steel.

On account of the expansion and contraction problem, open-type ball bearings, permitting longitudinal motion, are used. Bearing *A* is locked in place. Bearing *B* is backed by a special spring washer *C*, which compensates for the end motion caused by expansion or contraction. Spring *C* bears against *D*. It is interesting to note that a total temperature difference of about 145 deg. F. will cause the spindle to expand and contract approximately 0.001 in. per in. of length between the two ball bearings.

The quill or housing *E* is made of heat-treated or normalized cast iron whose inside diameter is ground to a permissible variation of 0.00015 in. Specially selected ball bearings whose S.A.E. limits have been halved are used. The quill is 8½-in. long with a tolerance of plus or minus 0.001 in. Care must be taken in establishing the tolerances so that the end caps will give the proper amount of clearance.

Lubrication at this high speed presented a problem which was overcome as follows: A long cylindrical felt tube *F* extends the entire length of the inner casting of the quill between the bearings. The ends of the cylindrical felt tube butt against the two felt washers *G* and *G*, these washers being designed to make contact with the tapered portion of the spindle at *H* and *H*, thus providing a continuous path for the oil to be conducted from the felt tube *F*, saturated with light oil, to the surface of the spindle. When the spindle is revolving there is a limited amount of heat generated under the surface of the felt washers *G* and *G*. This warmth decreases the viscosity of the oil, thereby increasing the capillary attraction which draws the oil from the long felt tube to the spindle. The oil accumulating on the surface of the rapidly rotating spindle is moved by centrifugal force over the surface of the spindle to the high point of the taper from whence it is thrown directly into the ball bearings in the form of a mist. Even with the spindle in continuous service, this type of lubrication requires oiling only once or twice a week. Labyrinth seals at each end prevent any appreciable loss of oil.

Sometimes end play in the spindle mounting is taken up by an adjusting nut which is set manually to give the desired loading on the ball bearings. Such a design is shown in Fig. 9 wherein the rotor of the driving motor is mounted directly on the spindle at *A*. The working load is comparatively light and hence the imposed axial load on the bearings need only be correspondingly small. This is adjusted by inserting a pin wrench through the slot *B* in the housing, engaging holes in the nut *C*. Turning the nut increases or decreases the "preloading" on the bearings.

In this design, one oil pocket, filled to the level *H*, supplies both bearings with lubricant. This oil pocket is cut into the stator housing *D* and the bearing housing *E* and is covered by a steel cover *F* which carries two hollow cut-away pins *G* and *G* containing the wicks. Drawn up by the wicks, the oil passes through the bearings and onto the slingers from which the excess lubricant is thrown off and through the holes *I* and *I* is returned to the reservoir.

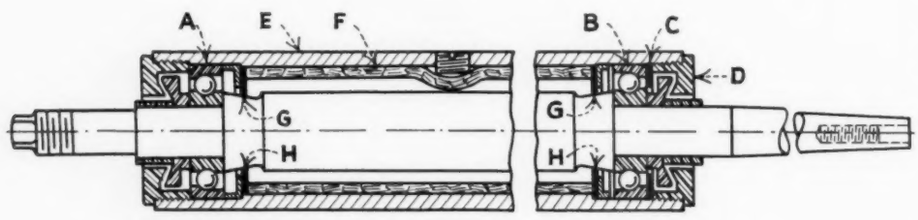



Fig. 8—Spindle mounting wherein provisions are made for expansion and contraction of spindle because of wide temperature variations

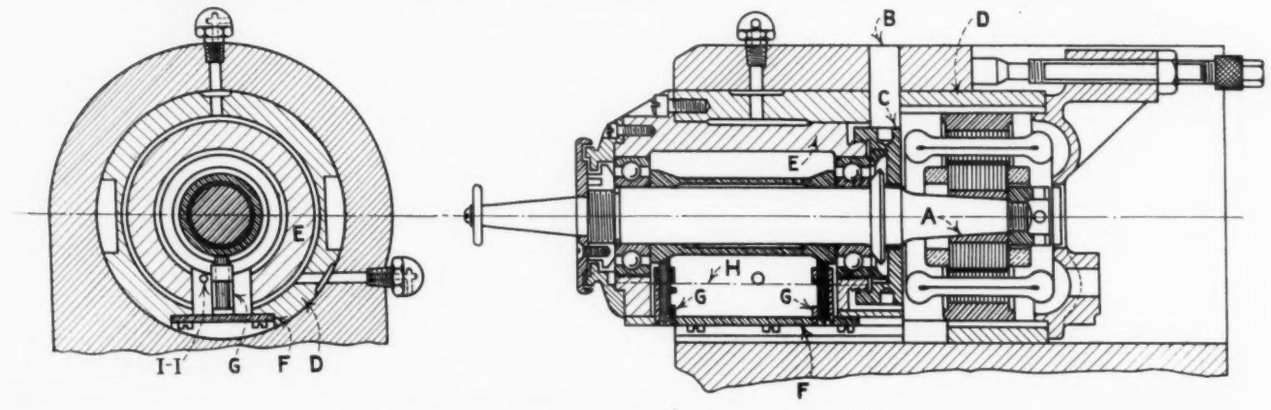



Fig. 9—A manually set adjusting nut to give desired loading on the ball bearing is used in this design

# PRODUCT DEVELOPMENTS

Welded diesel engines for high-speed trains—New cotton feeding system for pickers—Automatic two-speed drive for spinning frames—Hammer attachment for power shovels—Photocell protection for oil burners—Air-conditioned automobile

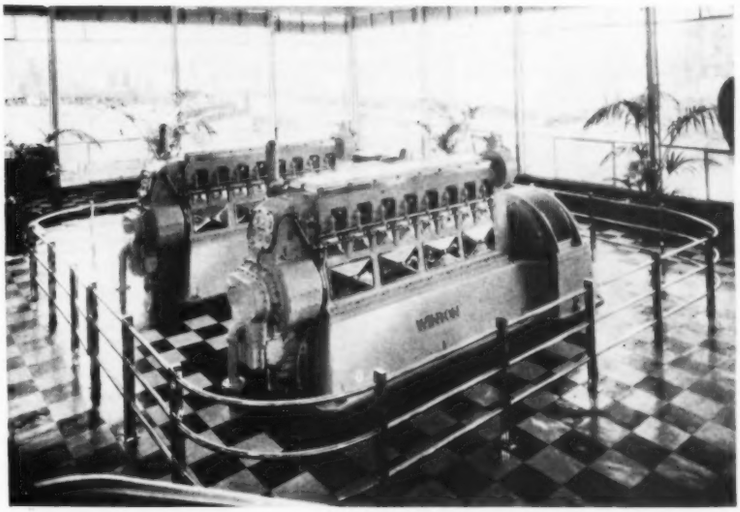
▼ ▼ ▼

## ▸ Welded Diesel Engine Crankcases

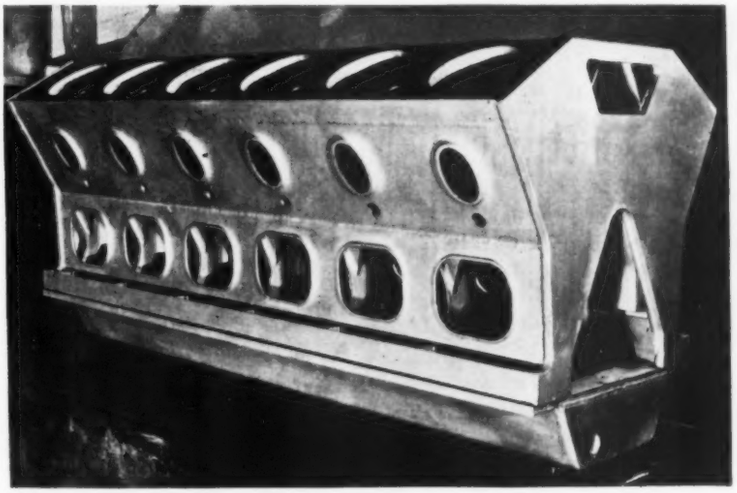
Many readers will recognize in the accompanying picture the diesel engines that form a part of the Winton Engine Corporation's exhibit at the Century of Progress in Chicago. These oil-electric units supply the power for the General Motors Exhibit Building. Not only do these engines represent a century of progress but this type of design is also establishing itself as the future gas-electric power plant for high-speed passenger-train service. This is because of the exceptionally low weight per horsepower, resulting from the use of welded construction for the crankcase of the engine.

The installations shown at the World's Fair have a specific weight ratio of only 39 lb. per hp., including the subbase and the Westinghouse generators. The specific weight ratio of the engine itself is only 20 lb. per hp. Each generator is rated at 500 kva., 4,000 volts, 3 phase, 60 cycle, 720 r.p.m., with 120-volt direct-connected exciters.

The design of the type of crankcase used in these Winton engines is shown in the accompanying illustration. It represents a joint development of the Winton Engine Corporation and Lukenweld Inc., a division of the Lukens Steel Company. This is the type of crankcase construction that will be used for the engine-generator units for the power plants of the new high-speed (110 mi. per hr.) passenger trains being built for the Union Pacific system. Also, the high-speed passenger trains that are being built for the Chicago, Burlington & Quincy will be powered by the same type of unit.



**Winton oil-electric units at the Century of Progress, which represent the latest in welded diesel engine construction**



**Type of welded crankcase design used by Winton, a joint development of Winton and Lukenweld**

A detailed description of the design and development, testing and fabrication of these welded diesel engine crankcases was presented by Everett Chapman, vice-president in charge of engineering, Lukenweld, Inc., before the Oil & Gas Power division of the American Society of Mechanical Engineers at their meeting held in Atlantic City on August

23. Mr. Chapman also described a welded steel crankcase for a 10-cylinder, 600-hp., radial engine for airplane service, in which engine the weight of the crankcase is less than 1 lb. per hp., which represents a distinct advance in diesel engine design.

## ▸ Cotton Feeder Breaks Tradition

When cotton pickers were fed by hand it was necessary to have a hopper on the floor into which the cotton could be fed manually. From this hopper the cotton was lifted by a vertical belt equipped with pins and the excess was pushed back by a beater or combing roll. This method prevented excess accumulations but was powerless to fill in the thin spots and an uneven mass of cotton was the result.

Even after mechanical feed supplanted manual feed the traditional hopper and feeder mechanism persisted, solely through tradition. Saco-Lowell Shops have introduced a new cotton-feeding system that breaks with tradition by abandoning the hopper feed and substituting a reserve chamber located above the rear of each picker and fed by a mechanical conveyor. In each chamber a delicately balanced feeler, connected to a mercoid switch, serves to keep the supply of cotton at a constant level and hence under constant pressure. At the bottom of the chamber a pair of large fluted rolls feeds a uniform sheet of cotton directly to the feed rolls of the picker.

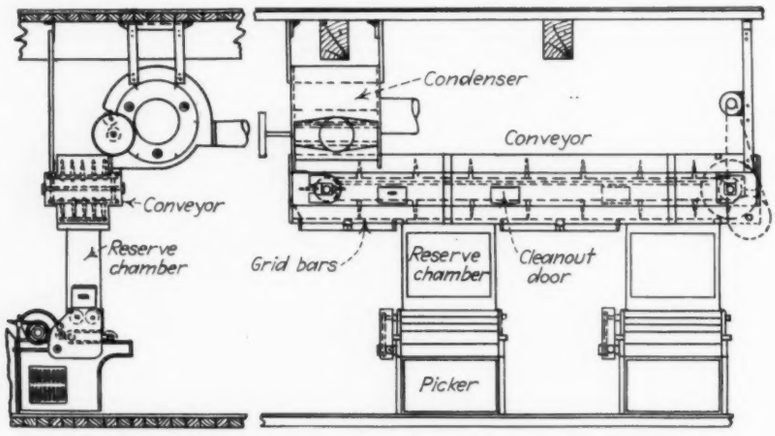
When all pickers have their proper reserve supply, the blending feeders and opening machinery are automatically stopped. When the level falls below a certain point in any one of the reserve chambers, the entire system starts up as a unit, cotton being available all along the conveyor for instant use. Two views of the machine, in section, are shown.

## ▸ Two-Speed Spinning Frame Drive

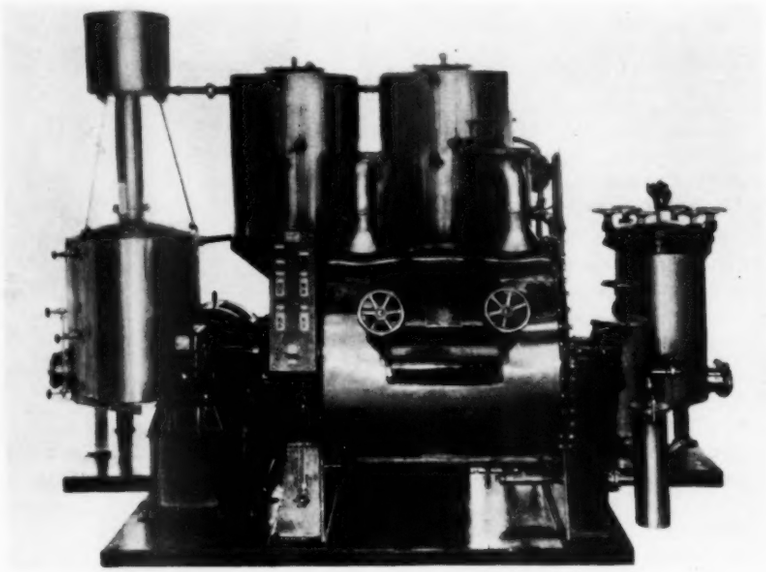
In the operation of spinning frames the tension in the yarns is a maximum at the start and is considerably reduced after a certain amount of yarn has been built on the bobbin. Because of the excessive tension at the beginning, it is necessary to spin during that period at a relatively low speed. In ordinary practice the spinning frame operates at a constant speed from the beginning to the end of the doff, hence the low starting speed is the determining factor in the production of the machine.

Whitin Machine Works now announces that in conjunction with the Westinghouse Electric & Manufacturing Company they have developed a two-speed drive which drives the spinning frame at slow speed for any desired period at the start of the doff and later shifts automatically to the desired higher speed. This is done by a limit switch cutting out resistance in series with the rotor windings. With such a drive it is necessary to spin only about 20 per cent of the yarn at the lower speed, while the speed during the spinning of the remaining 80 per cent of the yarn may be 10 to 15 per cent higher than the constant-speed drive when using non-variable speed motors.

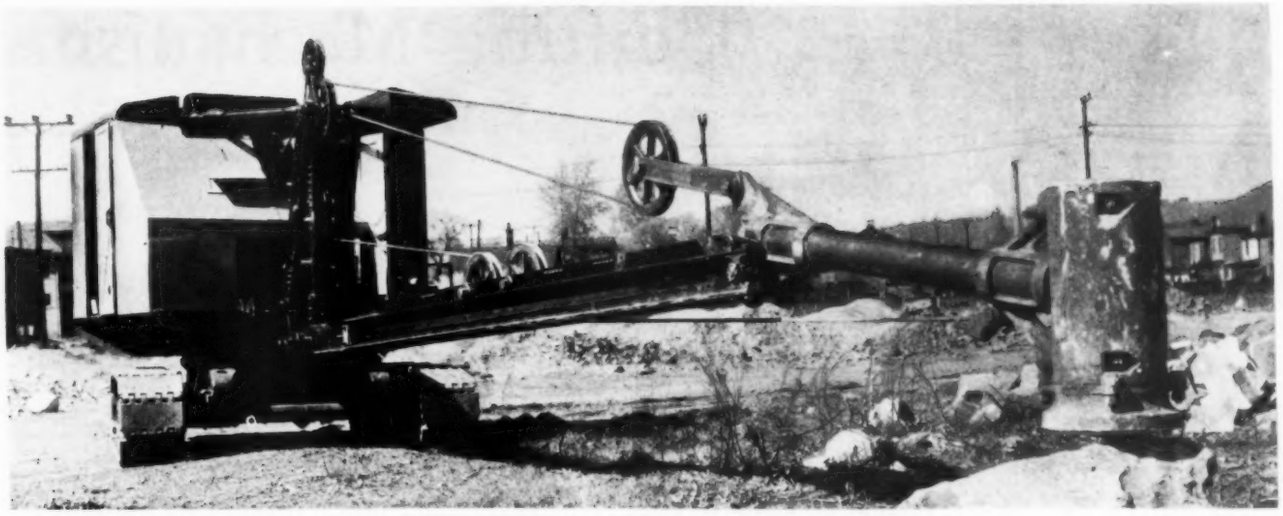
In this two-speed spinning frame drive, the motor can be stopped and started at any point within the doff without end breakage or strain on parts of the machine frame, since the magnetic control is arranged to always start the motor at reduced torque. If the motor is started dur-




**Longitudinal and cross-section views of the new blending, feeding and distributing system for pickers, developed by the Saco-Lowell Shops**



**Dry-cleaning equipment manufactured almost entirely of welded nickel by the Youngstown Welding & Engineering Company for the Safety Systems Company**



Removable hammer head weighing 3,000 lb. attached to the end of a steam shovel boom by an 8-ft. tubular handle. To the right is shown the hammer mounting for pushing, pulling or turning over large objects

ing the high-speed portion of the doff the changeover from low to high speed is controlled by a time-delay relay. By adjusting the resistor taps in the control cabinet, the preset slow speed can be adjusted to any value down to 30 per cent below the high speed. At no time is the operator required to change the speed through manual control.

### ▸ Hammer Attachment for Power Shovels

Attachments for power shovels are becoming as numerous as attachments for sewing machines. The latest is a 3,000-lb. hammer that can be attached to the boom end by an 8-ft. tubular handle, a development of the Keystone Driller Company. It is useful for breaking concrete pavement, ice or frozen earth crusts, castings, shale and sandstone bodies, oversize rocks and other obstructions; for demolishing small structures; for tamping and driving short piles or fence posts.

The hammer head has a removable striking face. Any desired shape for a particular job may be used, such as axe-shaped. Planetary clutches facilitate the operation of these tools, because they can be combined with clutches and brakes without excessive overheating. The hammer will deliver 20 blows a minute with a drop of 5 to 10 ft. It is so mounted that the hoisting line catches the flexible joint before the head hits the ground, thus simulating the action of a band hammer and freeing the joint from any appreciable strain.

### ▸ Photocell Protection for Oil Burners

One of the hazards that must be guarded against in the operation of oil burners is the accidental extinguishing of the flame. This results in an overflow of oil which might cause an explosion when subsequently ignited.

As is often the case, accessories provided by the user point the way to possible changes in design. At the plant of the Procter & Gamble Company, a dry-disk type photocell was mounted in a pipe leading directly into the firebox, the device being far enough removed from the furnace to prevent the cell from being damaged by heat. When the oil flame is extinguished and light no longer falls on the cell, the cell opens its sensitive relay which in turn opens a power relay. Thus the starter coil is de-energized allowing the starter to drop to the stop position, thereby shutting down the blower motor and closing the electrically operated oil valve that controls the flow of oil to the burner, effectively preventing an explosion.

The principal advantage of the photocell installation is that it is practically instantaneous in its operation not allowing oil to overflow. Any breakdown in the photo-electric

control that might prevent its operation will cause the oil supply valve to remain closed.

### ▸ Air-Conditioned Automobile

Principles established through experience with building and railroad car air conditioning have been applied to the automobile. An experimental installation made by the Campbell Metal Window Corporation, a division of the American Radiator Corporation, was demonstrated in New York recently.

Filtered air passes over a coil on its way into the car body. The coil cools the air in summer and warms it in winter. Sealing of the body helps to keep out traffic noise. The filter eliminates dust and exhaust fumes.

In the experimental model the refrigerating unit is driven off the engine but arrangements for drive from a special battery are under consideration.

# Shifting Mechanisms

## Typical linkages for the axial movement

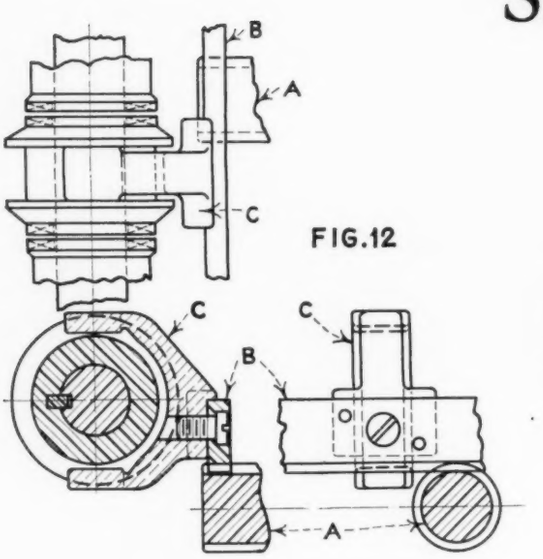



FIG.12

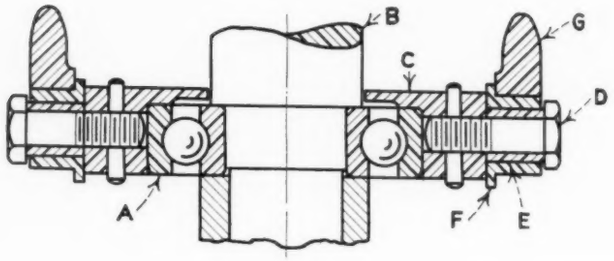



FIG.15

ADAM FREDERICKS

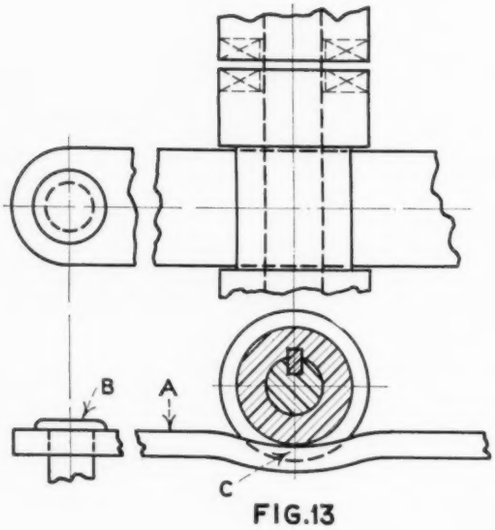



FIG.13

**Fig. 12**—The clutch is actuated by pinion *A* which is moved either by hand or power, engaging rack *B*, sliding in a groove of the same contour. Shifting fork C is screwed and dowled to the rack slide.

**Fig. 13**—An inexpensive clutch shifting arrangement wherein a shifting bar *A* of flat stock is pivoted at *B* and engages the clutch groove at *C*. Actuation of the clutch is done at the other end of the shifting bar *A*.

**Fig. 14**—Clutch shifting lever *A* is pivoted at *B* and actuated at *C* by the sliding bar *D*. Bushing *E*, a forced fit in yoke rack *F*, serves as both the seat for the shifting lever at *G* and the clutch locking means. Spring-backed ball *H* drops into hemispherical stop *I*. Shaft bearings *J* and *K* are capped.

**Fig. 15**—Bearing *A* is mounted solidly on the longitudinally movable shaft *B*. Housing *C* has two built-up trunnions. Screw *D* holding the spacing bushing *E* is mounted and pinned fast in housing *C*. Flange bushing *F*, free to rotate on bushing *E*, is engaged by the clutch releasing fork *G*.

**Fig. 16**—Two shifting levers *A* and *A* are keyed and screwed to shifting shaft *B*. Pieces *C* and *C*, similar to standard forged rod ends, are free to slide and accommodate trunnions of the solid yoke *D*. This yoke is held in place by flange *E*, screwed to gear *F*.

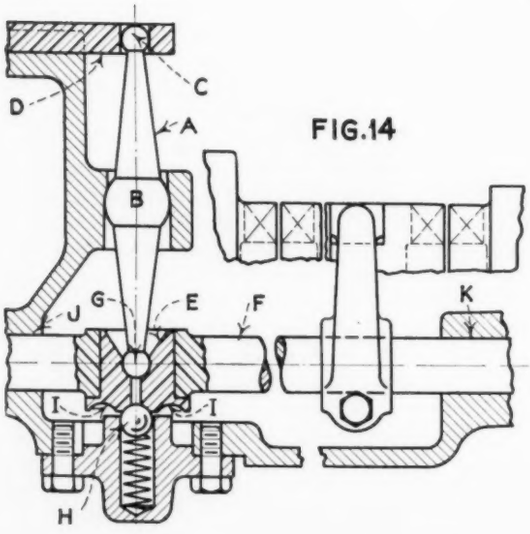



FIG.14

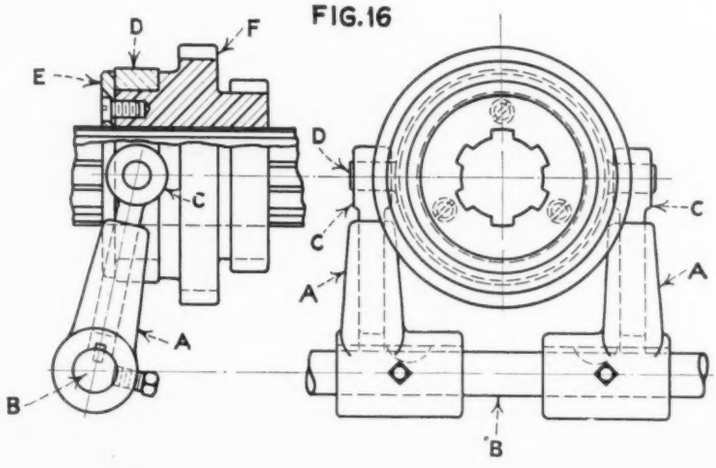



FIG.16

# for Gears and Clutches—II

of gears, clutches, wheels or other machine members

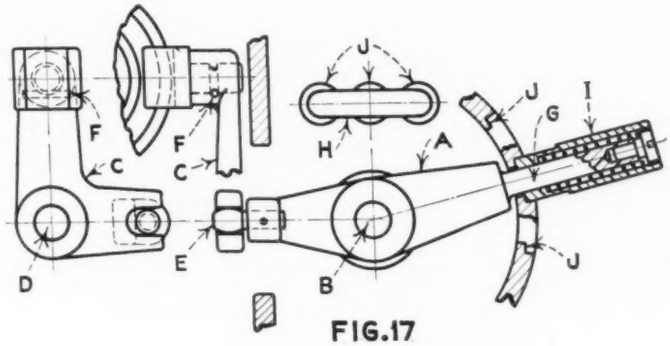



FIG.17

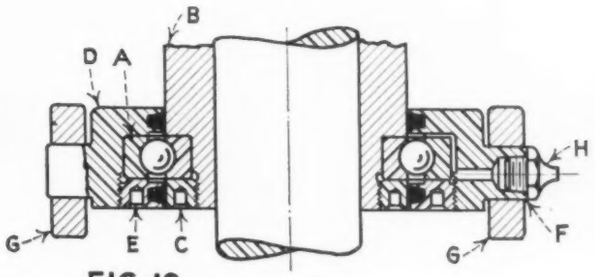



FIG.19

**Fig. 17**—Shifting lever *A* is pivoted at *B* and moves bell crank *C* which is pivoted at *D* by ball-ended pin *E*. Pin *F* retains shifting shoe and allows a slight rotation of the shoe when the gears are shifted. Projection *G* slides freely in slot *H* and end of knurled handle *I* is locked in the counterbored hole *J*.

**Fig. 18**—A hand-reverse mechanism for tooth clutches. Hand lever *A* attached to shaft *B* carries cam *C* which engages screw *E* in shifter rod *F*. Power drive is through shaft *H* keyed to sliding clutch *I*. Miter gears *J* and *J* run freely on bushings *K* and *K*. At *L* is the driven shaft. All views are shown in the neutral position.

**Fig. 19**—A two-direction ball-bearing clutch shifting mechanism. Bearing *A*, mounted on clutch member *B*, is held by nut C. Bearing housing *D* is held on by nut *E* and has two trunnions *F* which engage shifting fork *G*. Grease is introduced at *H*.

**Fig. 20**—For light-duty gears, the shifting shoe is clamped to shaft *B*. It engages the small space between the two gears *C*. The gears are positioned by the spring-backed ball *D* engaging the grooves *E*.

**Fig. 21**—A collapsible handle lock is used to position the gears. Shifting lever *A*, keyed to shifting rod *B* has mounted between lugs *C* and *C*, the lever *D* which actuates the locking plunger *E*.

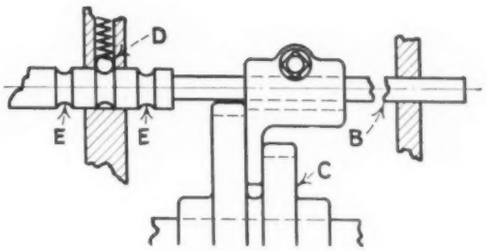



FIG.20

FIG.18

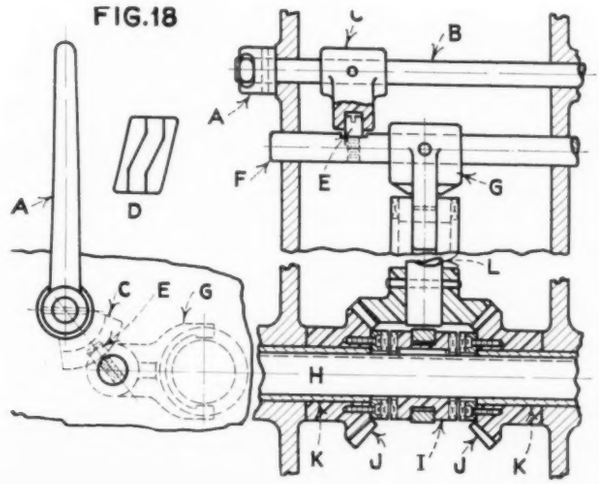
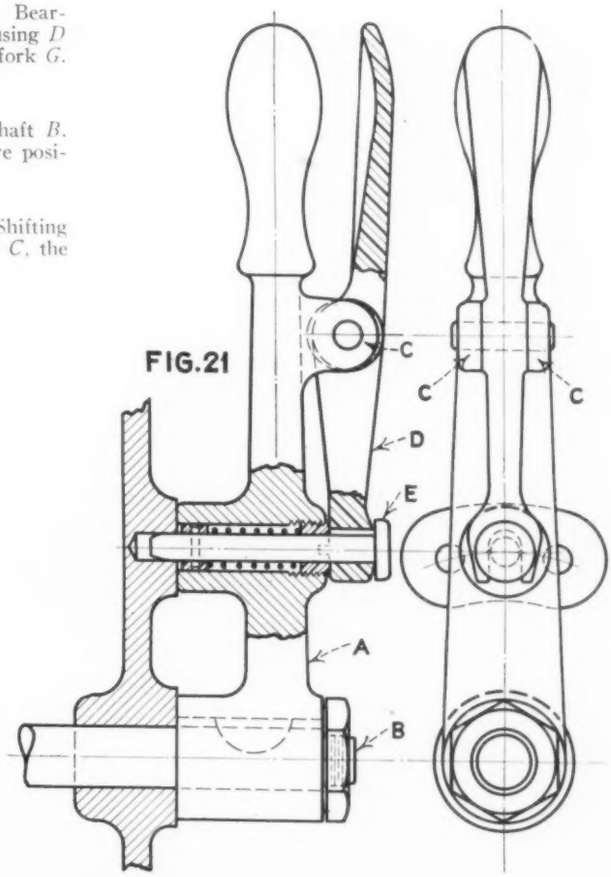
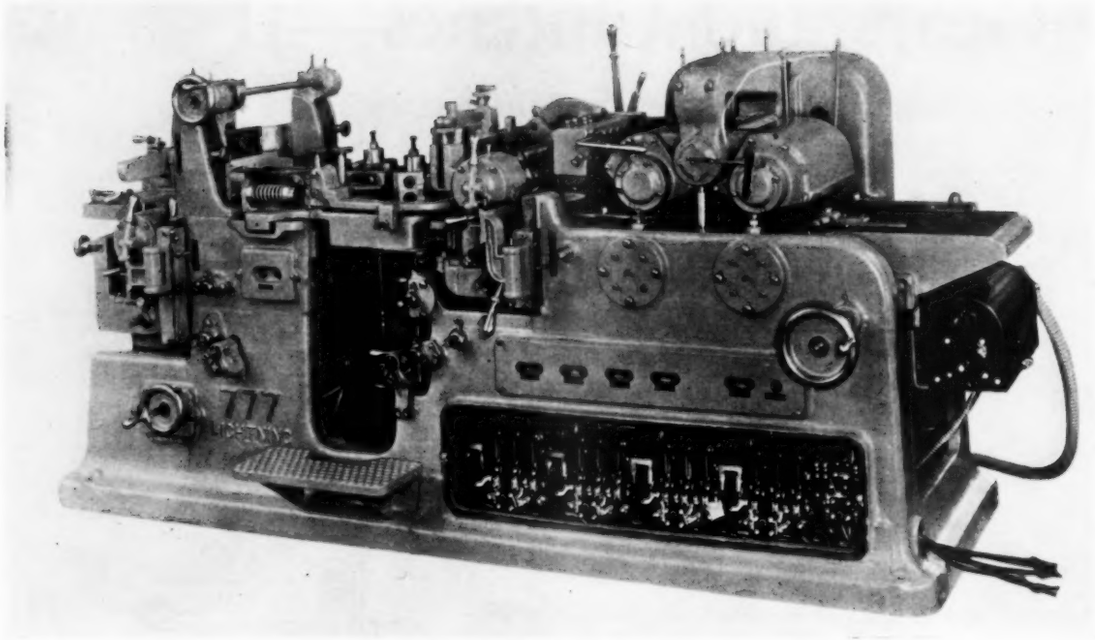



FIG.21





All the controls for this woodworking molder are mounted on a single panel and housed in the base of the machine

# The Control and Protection of Motors—I

## First of a series of articles on the construction and application of automatic starters and protective devices for electric motors

Motor controller functions are many, as can be seen in the accompanying tabulation. But there is one phase of the question that is general, namely, protection of equipment. Protection may be against over-heating because of a continuous overload or a locked rotor; against torque; against too rapid acceleration; or against restarting after a failure of the voltage or an overload. There are five general types of overload protective devices: fuses, thermal cutouts, thermal overload relays, magnetic overload relays and circuit breakers.

Fuses are the cheapest form of motor protection and also the least adequate. Their chief use is for tying-in with other protective devices to take care of short circuits. But fuses cannot always be used. For example, fuses cannot protect a polyphase motor. If a fuse in one phase blows, the motor may continue to operate single phase and thereby be in danger of burning out. With relays, on the other hand, if one relay operates, the motor is entirely disconnected from the line.

Thermal cutouts, like fuses, depend for their operation upon a fusible link of metal of low melting temperature. As in a fuse, there is a time delay in the operation of the thermal cutout inversely proportional to the amount of current overload. Heating of the fusible link is by the passage of current through a small heater coil surrounding it. Herein the operation of the thermal cutout is entirely different from that of a fuse in which the current flowing through the link itself does the heating. This difference gives the cutout a current-temperature curve that can be made to conform closely with that of the motor and makes it possible to give adequate protection without too frequent interruption of the circuit. Easily replaceable fusible links reduce the problem of maintenance. This type of motor protection is practically obsolete, having been replaced by thermal relays.

Thermal overload relays are also designed with a fusible element so that the heating of the fusible element follows closely that of the motor with which the relay is used. But no current flows through the fusible element. Melting of the alloy by means of a heater element releases a contactor in the control circuit, thereby disconnecting the motor. With this type of protective device, the circuit is not opened until the

motor has reached some predetermined temperature, regardless of whether this temperature has been produced by a large overload of short duration or a long continued small overload. Individual small overloads that do not raise the motor temperature to a point where the motor insulation is in danger, will not cause the relay to trip. But after the relay has tripped, time must be allowed for the fusible element to cool and solidify before the relay can be reset manually. This usually impels the machine operator to investigate the conditions of overload.

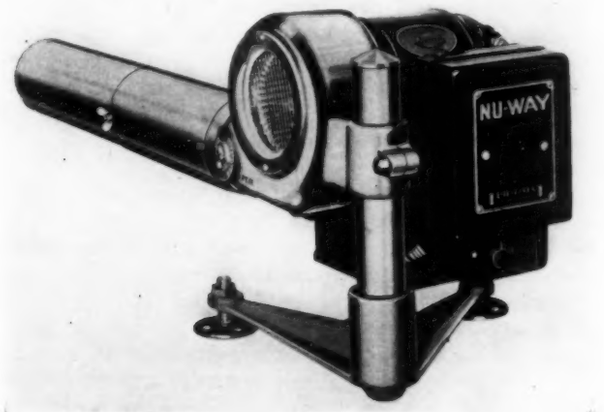
One type of thermal overload relay is built around a ratchet element imbedded in a low-melting alloy such as a eutectoid solder. When the solder melts a pawl is released and the contact is tripped. Heat may be applied to the elements either through heating coils, which is the usual method, or by means of a copper tube heated by induced current from a surrounding coil. In this design an adjustable iron core permits adjusting the flux passing through the copper tube and hence the time of action of the relay.

There are a few conditions, however, where a thermal overload relay acts too slowly to give adequate protection for a motor, in which case a magnetic or plunger-type overload relay is necessary. It is the only form of protection adequate to prevent torque overload on a d.c. motor, since it operates almost instantly. For example, before a thermal overload relay could act, a gear drive could be stripped. Magnetic relays can also give time delay inversely proportional to the current, by having the plunger operate in an oil dash pot.

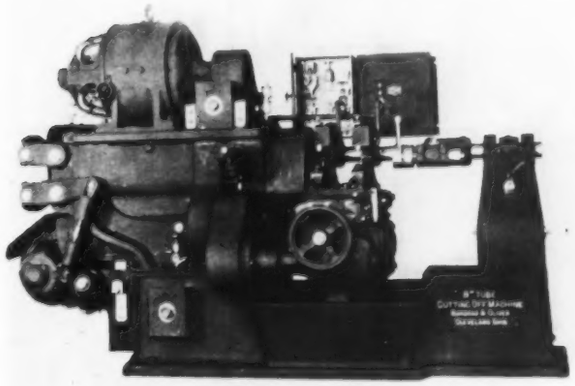
In the other type of thermal overload relay a portion of the current of the motor circuit is shunted through a bi-metal element. If heated above a predetermined temperature the bi-metal strip will bend with a snap action, opening the control circuit and thereby opening the contactors. This type of thermal overload relay also gives a time delay inversely proportional to the current.

It is obvious that for many types of loads a single relay will not give adequate protection and that a combination of the two is desirable. Furthermore, there are certain types of motors for which no adequate protection is available. A magnetic relay may be used to protect a repulsion-induction motor against locked rotor condition, and a thermal overload relay added for protection against a running overload. There is one condition in repulsion-induction motor operation, however, for which there is no protection; that is, when the motor slows down and the brushes drop upon the armature with resulting decrease in current, although the torque increases. Once started, the brushes begin to see-saw on and off the armature until destruction. If use could be made of current in the brush circuit, this condition could be stopped. For this purpose, it would be necessary to insulate two of the brushes, but so far motor manufacturers have resisted the change.

No-voltage or low-voltage protection devices vary




**By careful design, the controls can be mounted to harmonize with the machine. The small case below the main control cabinet contains an overload relay**



**In this tube cutting-off machine the control box is simply mounted in a convenient place. Latest practice is to house the controls in the frame of the machine**

**Ten Things that Motor Controllers Can Do Automatically**

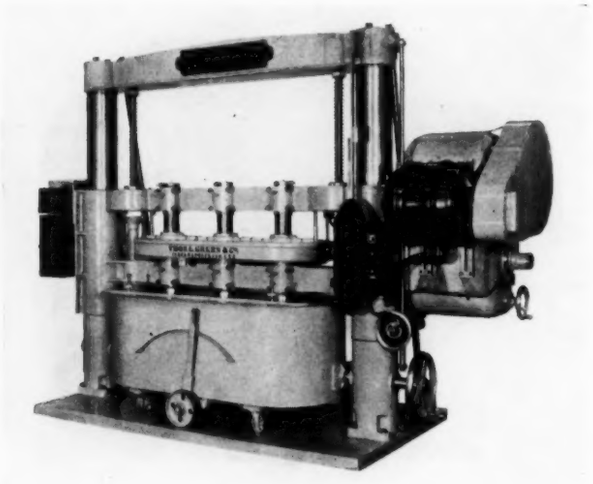
1. Connect motor to source of power
2. Reduce current inrush
   - (a) By use of resistors
   - (b) Auto-transformer (a.c. only)
3. Reverse motor
4. Provide dynamic or plugging braking
5. Speed control
   - (a) By insertion of resistance
   - (b) Interchange stator connections for multi-pole a.c. motors
   - (c) Limit light-load speed
6. Provide protection
   - (a) Low-voltage release (restarts itself)
   - (b) Low-voltage protection (manual reset)
   - (c) Overload release { Sudden overload / Overheating
   - (d) Overload protection
7. Apply mechanical brakes with operation of overload or low-voltage relays
8. Provide interlocking of controls for operation in sequence or to prevent improper sequence
9. Synchronize the speeds of two or more motors
10. Compel predetermined minimum time intervals between successive starts

according to the type of controller. For face-plate starters, either a.c. or d.c., an electromagnet coil connected across the line holds a latch in engagement and maintains the handle in the full-on position until voltage fails, at which point the latch releases and a spring returns the handle to its neutral position.

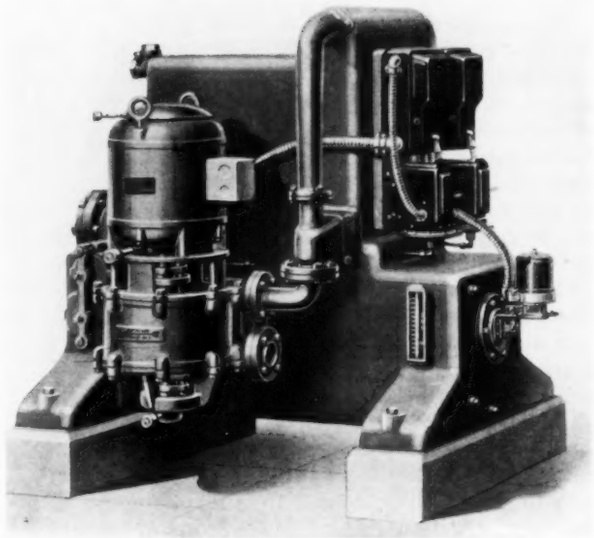
Automatic starters inherently provide under-voltage protection. When the start button is depressed, it energizes the coil of the main contactor and closes the main circuit to the motor. Usually, through a mechanical interlock, a second set of relay contacts are closed so that the circuit is maintained upon release of finger pressure from the start button. (In controls for jogging the motor, this interlock and secondary relay are omitted.) Should the stop button be depressed or should voltage failure take place on the line, the relay releases both the main contactor and the auxiliary interlocking contactor, making it necessary to depress the starter button before the motor can start up again. Using the same principle, no-voltage protection can be built into a master drum controller also.

Consumer products driven by fractional horsepower motors usually are manufactured in sufficient quantity to warrant careful study in the selection of controller equipment. Over 50 per cent of fractional-horsepower control equipment is special. In order to select intelligently a controller for this type of service, the controller manufacturer should know the horsepower and type of motor to be used, the current consumed in the locked position and the estimated time the motor may remain in this condition, the duty cycle. the surrounding temperature, the maximum load and length of application. On these jobs, tests are warranted on the complete machine to determine the soundness of the original selection.

Fractional horsepower motors, whether d.c., a.c. or of the universal type, require three forms of protection: against overload, locked rotor condition, and low voltage. A thermal type of relay serves these purposes. It may incorporate a bi-metallic element, but a simpler device as far as number of parts is concerned consists



**Automatic controls on a dough mixer, with the control equipment built in. Control devices include six limit switches and a 3-button push button station for controlling the reversing arm elevating motor**



**Ames vacuum heating pump equipped with: (1) a float relay switch for shutting down the motor when low water level is reached, (2) vacuum diaphragm relay switches to start motor at low vacuum and stop at high vacuum, (3) two across-the-line starters**

of a locked pawl and ratchet set in a eutectoid solder on the end of a copper stud about which the heater coil is wound. Such a cutout has all the characteristics of the larger thermal overload relays and will take care of a continued overload of a small amount or a sudden overload or current inrush, caused by a locked rotor condition. Under such circumstances the breaker will operate in 10 seconds, in sufficient time to save the motor insulation. As soon as the solder refreezes, the relay can be reset by hand.

The design of relay relative to speed of tripping is selected according to the type of service. Where the starting period is short, and where the stalled motor current does not exceed six times normal, a fast tripping heater relay can be used to save the fuses, which are necessary to protect against a dead short circuit. Fast tripping heater relays would be used on such applications as washing machines, fans, centrifugal pumps, meat slicers and power sewing machines. Where the motor must start heavy loads and the starting time is exceptionally long, a relay designed for slow tripping is used. But the fuse should be limited to not less than four times the ampere rating of the heating coils. Such a design of relay is used on such applications as grinders, compressors, refrigerators and reciprocating pumps.

Where a load can be started under reduced voltage, such as a fan or centrifugal pump, and where the starting current inrush must be held to a limited amount, a resistor is inserted in the motor circuit at starting. In one type of starting switch, an electromagnet in series with the motor and resistance holds an auxiliary contact open during the initial starting current inrush, but releases this auxiliary set of contacts as soon as the current drops to normal, thus short-circuiting the resistor and connecting the motor directly across the line through a primary set of contacts.

# Case Histories in Product Design—VI

T. J. MALONEY
*New Jersey Zinc Company*
and
GEORGE SWITZER
*Product Designer*

IN THIS closing installment of the present series of Case Histories, two striking developments in the realm of transportation vehicles are described. The first, case 17, relates to that now famous juvenile vehicle, the scooter car, which was merely a pair of roller skates nailed to a convenient board with an upright stick to steady the rider. Its ultimate development is shown here.

The second shown here, case 18, is perhaps a peek into the future. The amazing results obtained in the design of the Dymaxion car by a complete departure from standard automotive practice gives a strong hint as to what may be the future trend in automobile design.

## Product Case History No. 17

PRODUCT—Skippy Scooter.
MANUFACTURER—American National Company.
DESIGNERS—Van Doren & Rideout, Inc.
DESCRIPTION—Cognizant of the popular appeal that modern airplanes and automobiles hold for children from five to eight years old, Van Doren & Rideout designed the new Skippy scooter in correct streamline form. The two wheels are inclosed in teardrop fenders, similar to those on the latest automobiles. Even the footboard is carefully streamlined. In every detail this principle is carried to its logical conclusion.

Appearance is not the only improvement. The new scooter is entirely practical. For example, the front fork of the scooter is a single metal stamping integral with the front-wheel fender. The front wheel projects just far enough in front of the fender to absorb the shocks from hitting a curb, a tree or any impeding object. The penetration of the upper end of the footboard riser at the kingbolt, into the front housing, preserves the streamline effect and provides an entirely new method of construction. The steering handle slopes back with the wind and gives a flashy appearance. At the same time, it prevents the child from leaning over too far and insures easier operation.

Thus, while giving the Skippy scooter genuine streamline, the designers were also responsible for many mechanical improvements, leading to greater speed and ease of operation.

SALES RESULTS—Since this product was primarily designed for the coming Christmas season, it is too early for any retail sales records resulting from the new design. However, the opinion of buyers who have already seen it is that it will sell itself on sight. This fulfills the original objective of the designers to plan a scooter that would be above competition. And the orders coming in at present to the manufacturer indicate just such a reception for the streamlined model. For example, it went into production on June 15 and it has proved to be the fastest selling scooter the company has ever had.

The old familiar "scooter bike" in its ultra-modern form. This new model copies the latest developments in wheel boots and streamlining

## Product Case History No. 18

PRODUCT—The Dymaxion Car.
MANUFACTURER—Four D Company, Inc.
DESIGNERS—Fuller and Burgess.

**The Dymaxion three-wheel car powered by a Ford V-8 engine. Because of the excellence of its streamlining, this ultra-modern automobile is capable of a speed of 120 mi. per hr., and can travel 40 mi. on a gallon of gas**

DESCRIPTION—The Dymaxion has three wheels, two in the front and one in the rear. Drive and braking is on the two front wheels which carry 75 per cent of the total weight. The rear wheel serves for steering and can be turned through 180 deg. making it feasible to turn the car completely around in a circle whose radius is equal to the wheelbase. This wheel arrangement makes possible the excellent streamlined wheel designed by Mr. Burgess. The car is driven by a Ford V-8 engine, the engine being placed in the rear but driving the front wheels. The brakes are on the front wheels only.

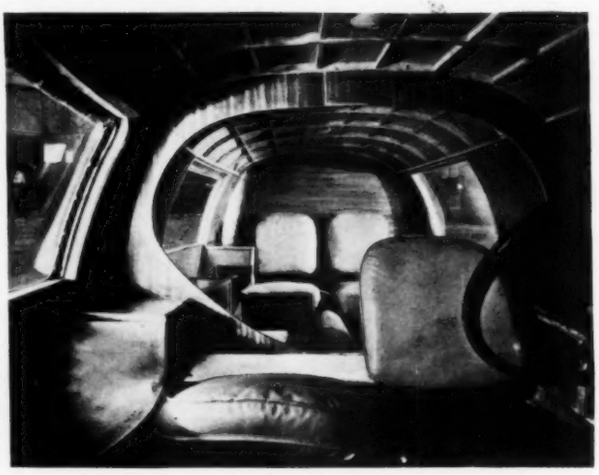
The Dymaxion weighs less than the Ford from which its motor came and, the designers claim, will travel forty miles on a gallon of gasoline and can attain a maximum speed of 120 miles per hour. Both the low fuel consumption and the high speed are attributed to the ultra-modern streamlininig whereby the wind resistance is reduced to a minimum. The fact that this car is powered by a standard Ford engine makes its performance all the more remarkable. Its overall length is 19 ft., with a wheelbase of 125 in. The volume of useful inclosed space is 378 cu.ft. as compared to approximately 143 cu.ft. in the present medium-price range standard sedan.

The interior is exceedingly roomy and can comfortably accommodate six people. There are no running boards and fenders. The headlights are in wells at the front, nothing breaking the smooth line of the exterior except a shallow scoop on the top for air to enter the radiator and a periscope sighting device for the driver, giving a much wider rear vision than the present automobile.

From a manufacturing point of view the Dymaxion is at present a custom-built and highly specialized job. The designers contend, however, that once the mass market appreciates the economy and feasibility of the car it will be produced in mass production at a lower price per pound than our present low-priced cars.

SALES RESULTS—Entirely problematical. The public is being presented with an automobile at least five years in advance of present automotive design. Great interest in the car has been evidenced by all automotive authorities and a surprising amount of interest—other than novelty interest—has been shown by those who have seen the actual demonstrations. The revolutionary design to be sold will take merchandising ability amounting to genius—even though a comparatively large number of individual orders, eighteen, have been received already. The car is now at the Chicago World's Fair.

It is the opinion of the manufacturers that there is a rapidly growing public appreciation of the value of streamlining. The advantages of the high speeds attainable combined with low fuel cost is daily becoming of greater significance.



**By virtue of its principles of body construction, the Dymaxion car has more than two and one-half times as much inclosed useful space as the average medium-price car**

# Types of Gears and Their Mountings

P. C. DAY

*Chief Engineer, The Falk Corporation*

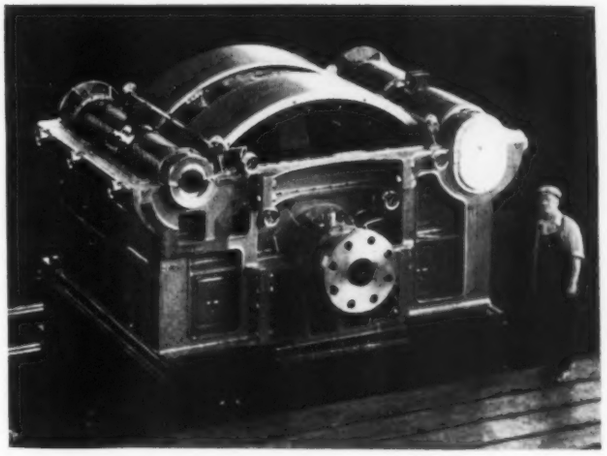
IN THE design of the conventional gear train, it is a question as to which of the three types of gearing, spur, herringbone or helical, should be adopted. The answer to this question depends entirely on the nature of the application and the requirements to be met. Power to be transmitted, the value attached to high efficiency, silence of operation, smooth running, compactness and inherent design qualities are the principal factors to be considered. These must be studied on the basis of cost versus market values. For rough cast gears such as are frequently used in construction and similar equipment, the straight spur-gear tooth is unquestionably the type to be used. The other extreme is turbine gearing wherein large power must be transmitted at high pitch line speed. For such applications helical gears are used exclusively. Generally speaking, as speed and load increase, either herringbone or helical type gears should be used. Also, when high efficiency, silence of operation and compactness are important considerations, the herringbone and helical type gears will be superior to the straight spur gears.

Regardless of the particular type and form of the teeth used in the gear train, the same fundamental principles control the design details. In all speed reducers it is a problem of torque multiplication rather than speed reduction. As in most engineering problems, forces and bending moments are the dominant factors.

As an example of the influence of torque multiplication in a gear train, one of the questions in the design of motor and speed reducer units is whether the motor shall support the reducer or vice versa. In considering this question, when standard motor frames are used the frame, feet, and holding-down bolts are designed on the basis of the maximum torque output of the motor. The reducer unit multiplies this torque almost in direct proportion to the speed reduction. For a given speed and torque output, as the reduction ratio increases the motor and the maximum motor torque become smaller. From this it follows that a point is soon reached where the size of the motor frame is insufficient to properly support the reaction of the torque delivered by the slow-speed output shaft.

It must also be considered that on account of the necessary overhang of the gear unit, the torque reaction from the output shaft of the motor reducer will result in large bending moments which are transmitted to the motor frame. Therefore, except for low reduction ratios the speed reducer should support the motor. Reaction of the torque on the motor frame will then be the delivered torque of the motor, regardless of the reduction ratio of the gear unit.

In comparing the different types of gears, ordinary commercial spur gears have a limited field of application with reference to load carrying capacity and pitch line speed. This follows from the fact that spur gears do not have continuous tooth contact at the point of tangency of the pitch circle. Throughout the whole of the arc of action, the number of pairs of teeth in contact simultaneously rarely exceeds two. With high gear ratios and a small number of pinion teeth, it is usually impossible to have more than two pairs of teeth in contact at any one time. For this reason, the smoothness of action of spur gears depends entirely on the accuracy of the tooth profile over the whole of its active portion. Inaccuracies at any point in the profile of the spur gear tooth are almost bound to result in vibration and noise with a lowering of the load carrying capacity of the gear. The accuracy required for smoothness of operation of spur gears can be obtained only at a cost which will make them almost as expensive as accurately formed herringbone teeth.



**One of the turbine gear units transmitting 27,750 hp. for driving the propellers of the new treaty cruiser Tuscaloosa. The helical gears are flood lubricated with pressure lubrication to all the bearings**

Because of the inherently smoother running of herringbone and helical type gears, which is reflected in an absence of vibration, such gears can be run at higher pitch speeds and higher velocity ratios than straight-tooth spur gears. Whereas the velocity at the pitch circle is commonly limited to 1,000 ft. per min. for straight spur gears made of high carbon steel forgings, helical gears of the same material and of a comparable design, can be run at a velocity of 12,000 ft. per min. at the pitch circle.

One of the reasons for the inherently smoother running of herringbone and helical type gears arises from the fact that slight inaccuracies in the tooth profile will

not necessarily be as detrimental as similar inaccuracies in the profile of straight spur gear teeth. Helical and herringbone gears always have one or more pairs of teeth in contact at the point of tangency of the pitch circle. Therefore, as long as there are no points on the tooth profile lying outside of the true profile, smooth running will be obtained.

Editor's Note—*For a detailed comparison of variation of length of tooth contact throughout one pitch rotation of straight spur herringbone and helical gears, see article "Tooth Contact in Spur and Helical Gears" by H. E. Kitchen in the August, 1932, Product Engineering.*

Considering the relative merits of the various tooth forms on the basis of strength in both herringbone teeth and helical teeth the line of contact across the face of the tooth is along diagonals. Also, because the teeth do not lie parallel to the axis of the gear, they are inherently stronger. Both of these factors contribute to making it possible to design herringbone or helical gears having a smaller diameter or a finer pitch than would be possible if straight spur gear teeth were used. This is further accentuated by the fact that with a greater number of teeth of finer pitch, a higher tooth-strength factor can be applied. Thus, for the same physical dimensions, helical gears can carry higher working loads because the tooth stresses will be lower.

With a finer pitch, smoother running is obtained, and the diameters of the gears can be made smaller. From this it follows that because the gears are smaller in diameter they can be run at a higher r.p.m. for the same velocity at the pitch circle.

Illustrating the great strength of helical type gearing, a turbine reduction unit designed recently has helical gears carrying a load of 1,000 lb. per in. of face width. The teeth of the gears are only $\frac{5}{16}$-in. thick, measured circumferentially at the pitch circle. The helix angle of the teeth is 30 deg. The pinion on the high-speed shaft is 9.8-in. pitch diameter and runs at 3,460 r.p.m. This reduction unit transmits 20,000 hp.

Contrary to general opinion, the permissible safe width of helical and herringbone gears is not far greater than that which is allowable for straight spur gears. The limiting factors on the face width of the gears are the bending moments and torsional moments to which the gears are subjected. These depend on the ratio between the length and diameter of the pinions. The capacity of any gear, regardless of the type of tooth profile used, varies in direct proportion to its face width, but it is also proportional to the square of the center distance between the gear shafts. On the other hand, for a given set of conditions, using gears of the same diameters and same pitch, the required width of helical or herringbone gears to transmit a given horsepower is considerably less than the width for spur gears. This can readily be seen from a comparison of the capacity of helical (or herringbone) gears with spur gears for a 10 hp., 5:1 speed reducer unit. For a motor speed of 1,750 r.p.m., for both types of gearing, a center distance of $4\frac{1}{2}$ in. would be chosen. Precisely the same materials would be used for both types of gearing, namely, the pinion an alloy steel forging, heat-treated 35/40 Shore and the gear 0.40 to 0.50 carbon steel forging, heat-treated to 30 Shore for both types of gears; the pitch would be the same, namely, 16 D.P., the pinions having 24 teeth. The correct face width for the helical or herringbone gears would be $1\frac{3}{4}$ in. while the corresponding face width for spur gears would be $2\frac{5}{8}$ in. Thus, the straight spur gear unit is 50 per cent wider than the similar unit using herringbone or helical gears.

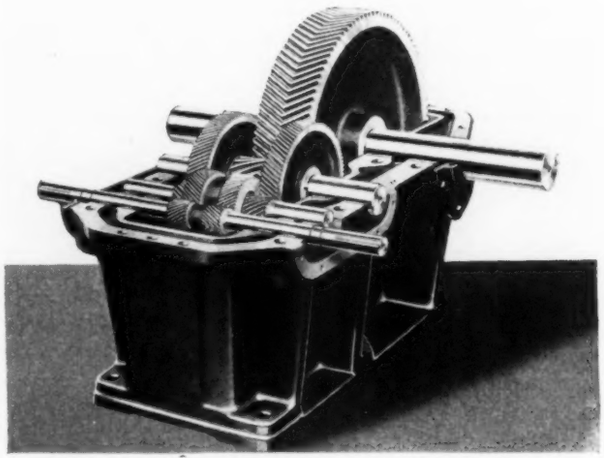
In the mounting of single helical gears it is necessary to make some provision for the axial thrust. But by mounting a right-hand and left-hand helical gear on the same shaft, the two thrusts will balance each other. With reference to thrust, a single herringbone gear is equivalent to a double helical gear.

One precaution that should be taken in the mounting of both double helical gears and herringbone gears is that one or the other of the gears be permitted to float axially. The V-shaped herringbone pinion tooth (or double helical tooth) must be permitted to wedge into its mating gear tooth so that the load will be divided equally between the two sides of the V. Any axial thrust in the mounting would indicate that there is not an equal division of the load. But unless prevented from doing so, one or the other of the mating gears will shift axially until an equal distribution of the pressure has been reached.

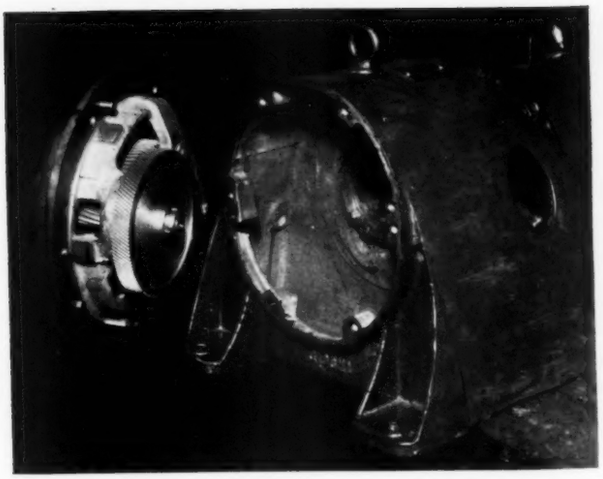
In a train of gears it is not always possible to arrange the construction so that one of each pair of mating gears will be permitted to float axially. Under such conditions, single helical gears are used for those pairs of gears wherein axial shifting cannot be permitted.

Having established the type of gearing to be used and the design of the gears, the next major question is the type and design of the bearings. Although this may be largely a matter of personal choice, there are certain conditions that will dictate either sleeve-type bearings or some form of anti-friction bearings.

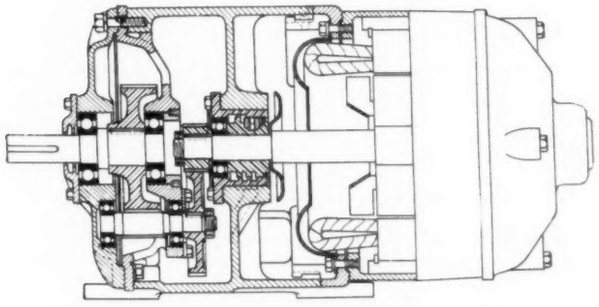
It is the practice of the Falk Corporation to prefer sleeve-type bearings for parallel shaft reducers wherein the design allows balanced moments and pressures on each of the two bearings supporting one shaft. Babbitt-lined bronze bearings are used; they are made in halves for convenience of assembly and replacement.



**A standard type triple reduction parallel shaft speed reducer with sleeve bearings, as manufactured by the Falk Corporation, shown with cover removed. Oil splashed on the inside of the cover drips into the groove in the top of the lower housing and is distributed to all the bearings**



**Special reduction gears mounted as a unit in a motor speed reducer, a type of design that simplifies inspection, maintenance and replacement to change the speed ratios. A sectional view of the complete unit is shown below**



On motor speed reducers other factors enter into the selection of the type of bearing. It is necessary to reduce the length of the bearings as much as possible in order to arrive at a compact construction. Also, symmetrical loading of the bearings is impossible because of the straight-line arrangement of the gears. For such construction, the Falk Corporation uses ball and roller bearings exclusively, but, at the same time, single helical gears are substituted for herringbone gears.

To illustrate the amount of room saved by the use of anti-friction bearings in preference to plain sleeve-type bearings, the accompanying table gives the length of the bearings for each of the two types of mountings for reducer units having approximately equal horsepower ratings.

Lubrication is of extreme importance. The design of any speed reducer should provide for a constant flow of oil to all bearings and a continuous supply to the gear teeth. Reducers which operate at medium speed are usually lubricated by the splash system through one or more gears being allowed to dip into a bath of oil which is stored in the lower part of the gearcase.

In the accompanying illustration is shown the design of gear housing used by the Falk Corporation in order to assure a flood of lubrication to all of the bearings. As can be seen in the picture, the upper face of the housing extends inside the cover so that oil flowing downward on the inside surface of the cover will be deposited on the inwardly extending surface of the lower housing. Grooves are cut in the top surface of the lower housing, these grooves conducting the oil to each of the bearings. At each of the bearings an oil passage is provided so that the oil from the groove can flow down under the shaft and up into the corresponding groove on the other side of the bearing. Thus there is provided a continuous channel for the lubricating oil to flow to each one of the bearings.

With reference to the efficiency of herringbone and helical type gear trains such as described here, single reduction units running at full load and normal speed will show an efficiency between 98 and 99 per cent. For double reduction units the efficiency ranges between 97 and 98 per cent while for triple reduction units the efficiency ranges between 96 and 97 per cent. These efficiencies are based on test results and are subject to changes that may be caused by operating conditions. For example, the viscosity of the lubricant will have some effect. For shock loads it is necessary to use an oil of high viscosity. This will decrease slightly the efficiency of the unit, but not to any marked extent.

One of the questions frequently asked with reference to gear speed-reduction units is the maximum gear ratio that should be employed. Technically, there is no limit, but in practice there are several limits which depend on the nature of the problem in hand.

A pinion of given diameter and running at a fixed speed under given conditions will transmit a definite load in horsepower per inch of face. However, the permissible "face" is limited by the diameter chosen. In general practice the face is limited to $2\frac{1}{2}$ diameters to avoid excessive bending and torsion.

In general, a given pinion will tranmit more power when working with a larger (high ratio) gear than with a smaller one. However, since the diameter of the gear is necessarily equal to the diameter of the pinion multiplied by the ratio, it follows that high single ratios lead to dimensions which are unsuitable and uneconomical.

In standardized commercial speed reducers of the independent (parallel shaft) type the practical limit for single reduction is about 10 to 1. The more compact "Motoreducer" motor reducer as manufactured by the Falk Corporation, has a top limit of 9 to 1 but ratios of 6 to 1 or less are more usual. Higher ratios require double or triple gear trains.

COMPARISON OF BEARING LENGTHS

Sleeve Type vs. Roller Bearings

| | No. 3 DA Falk speed reducer with sleeve bearings | No. 28 Motor reducer with roller bearings |
|---|---|---|
| High-speed shaft | 1.700 in. | 0.590-0.630 |
| Intermediate shaft | 2.000 in. | 0.669-0.748 |
| Low-speed shaft | 3.000 in. | 0.827-0.787 |
| | **No. 7DA Falk reducer with sleeve bearings** | **No. 56 Motor reducer with roller bearings** |
| High-speed shaft | 3 in. | 0.827-1.221 |
| Intermediate shaft | 4 in. | 1.625-2.000 |
| Low-speed shaft | 5 in. | 1.750-2.125 |

# Classification and Characteristics of th

*Contributed by T. Holland Nelson, Consulting*

| GROUP CLASSIFICATION Typical Composition of Each Type in Per Cent | | | | | | | MECHANICAL PROPERTIES AT NORMAL TEMPERATURES (See Note 1) | | | | | | | ULTIMATE AND WORKING STRESSES AT ELEVATED TEMPERATURES Temperatures in Deg. F. (See Note 2) | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Type | C | Si | Mn | S & P | Cr | Ni | Yield Point | Ultimate Stress | Elong., Per Cent in 2 in. | Red. of Area Per Cent | Impact ft. lbs. | Hardness Brinell | Hardness Rockwell | 1,200 | 1,400 | 1,600 | 1,800 | 2,000 |
| A-1 | 0.07 | 0.08 | 0.12 | 0.025 | 11.70 | Tr | 60,000<br>140,000 | 80,000<br>160,000 | 30<br>10 | 70<br>35 | 90<br>30 | 170<br>340 | 85-B<br>35-C | 24,000<br>3,600 | 12,000<br>1,560 | 10,000<br>1,025 | ..... | ..... |
| | 0.35 | 0.19 | 0.15 | 0.025 | 12.00 | Tr | 100,000<br>180,000 | 110,000<br>240,000 | 25<br>5 | 55<br>20 | 35<br>5 | 220<br>440 | 20-C<br>45-C | ..... | ..... | ..... | ..... | ..... |
| A-2 | 0.09 | 1.25 | 0.35 | 0.025 | 18.50 | Nil | 50,000<br>80,000 | 75,000<br>95,000 | 35<br>25 | 75<br>60 | 0<br>12 | 140<br>200 | 80-B<br>95-B | ..... | ..... | ..... | ..... | ..... |
| A-3 | 0.23 | 0.75 | 0.65 | 0.025 | 28.50 | Tr | 60,000<br>80,000 | 80,000<br>120,000 | 35<br>25 | 60<br>40 | 6<br>2 | 150<br>210 | 80-B<br>95-B | 40,000<br>6,000 | 18,500<br>2,400 | 9,000<br>925 | 6,500<br>325 | 5,000<br>120 |

Castings from the above alloys have elongation and reduction in area from 5 to 20 per cent, much less t

| | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| B-1 | 0.08 | 0.25 | 0.40 | 0.03 | Nil | 5.00 | 35,000<br>65,000 | 75,000<br>100,000 | 35<br>25 | 70<br>55 | 90<br>50 | 160<br>210 | 85-B<br>95-B | ..... | ..... | ..... | ..... | ..... |
| B-2 | 0.15 | 0.20 | 0.43 | 0.03 | Tr | 36.10 | 30,000<br>45,000 | 70,000<br>90,000 | 35<br>25 | 65<br>55 | 110<br>60 | 160<br>170 | 85-B<br>85-B | ..... | ..... | ..... | ..... | ..... |

B-1 material, resistant to mild forms of corrosion, was extensively used for turbine blades; has recent
B-2 material is used where an austenitic material is required. This type of material with 10 to 12 per

| | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| C-1 | 0.07 | 0.30 | 0.40 | 0.03 | 18.00 | 8.00 | 30,000<br>45,000 | 80,000<br>100,000 | 65<br>50 | 75<br>65 | 120<br>90 | 130<br>160 | 75-B<br>85-B | ..... | ..... | ..... | ..... | ..... |
| | 0.15 | 0.30 | 0.40 | 0.03 | 18.00 | 8.00 | 30,000<br>45,000 | 80,000<br>100,000 | 65<br>50 | 75<br>65 | 120<br>90 | 130<br>160 | 75-B<br>85-B | Brittle Range | 30,000<br>3,900 | 15,000<br>1,525 | 9,000<br>450 | ..... |
| C-2 | 0.25 | 0.35 | 0.45 | 0.03 | 25.00 | 10.00 | 40,000<br>60,000 | 80,000<br>110,000 | 55<br>35 | 70<br>50 | 90<br>50 | 150<br>200 | 80-B<br>95-B | 55,000<br>8,250 | 34,800<br>4,520 | 18,250<br>1,870 | 10,850<br>540 | 5,500<br>125 |
| C-3 | 0.15 | 0.75 | 0.60 | 0.03 | 25.80 | 19.75 | 40,000<br>45,000 | 80,000<br>95,000 | 50<br>30 | 65<br>50 | 100<br>60 | 140<br>180 | 80-B<br>90-B | 50,000<br>7,500 | 35,000<br>4,550 | 20,000<br>2,000 | 11,500<br>575 | 6,500<br>150 |

High silicon, copper or molybdenum are often found in the C-group. Titanium, vanadium and c
precipitation. Selenium or zirconium sulphide may be added for free machining. Where some of th

| | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| D-1 | 0.15 | 0.50 to 2.00 | 0.45 | 0.03 | 10.00 | 20.00 | 45,000<br>60,000 | 85,000<br>100,000 | 35<br>20 | 55<br>35 | 90<br>50 | 160<br>200 | 85-B<br>95-B | 52,000<br>7,800 | 32,000<br>4,150 | 16,000<br>1,625 | ..... | ..... |
| D-2 | 0.35 | ..... | ..... | 0.03 | 12.00 to 20.00 | 35.00 | 45,000<br>65,000 | 80,000<br>110,000 | 35<br>20 | 60<br>45 | 60<br>40 | 160<br>180 | 85-B<br>90-B | 55,000<br>8,250 | 40,600<br>5,275 | 23,400<br>2,400 | 13,250<br>650 | 7,500<br>175 |
| D-3 | 0.45 | W 2.00 | .... | 0.03 | 20.00 | 60.00 | 35,000<br>50,000 | 60,000<br>80,000 | 35<br>20 | 60<br>45 | 100<br>50 | 140<br>160 | 80-B<br>85-B | 55,000<br>8,250 | 35,250<br>4,575 | 22,500<br>2,800 | 15,250<br>750 | 9,250<br>225 |

D-1 type is obtainable in nearly all forms including seamless pierced and drawn tubes. D-2 type with mod
For turbine blading a lower chromium content is used for temperatures above 800 deg. F. D-3 type is
forgings although it is difficult to fabricate. With the addition of 15 to 20 per cent molybdenum this mat

Note 1—In columns headed yield point and ultimate stress the first figure refers to the annealed condition; the second figure is for cold worked or hardened material. Differences between these figures and other published data are accounted for by modifications of analysis or by variations in heat-treatment or work hardening during fabrication.

# of the Iron-Nickel-Chromium Alloys

*, Consulting Metallurgist, Philadelphia, Pa.*

| G STRESSES RATURES (See Note 2) | | AVAILABLE FORMS | | | | | | | FORGING, HARDENING AND ANNEALING TEMPERATURES Temp. Range, Deg. F. | | | GENERAL INFORMATION | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1,800 | 2,000 | Castings | Forgings | Billets | Bars | Cold dr. or Grnd. | Sheets, Plate, Stp. | Seamless Tubes | Forge | Harden | Anneal | Notations | Applications |
| .... | .... | √ | √ | √ | √ | √ | √ Seldom used | √ | 2,100<br>1,600 | 1,750<br>1,800 | 1,550<br>1,600 | Magnetic<br>Hardening Type | General engineering purposes, turbine blading. |
| .... | .... | √ | √ | √ | √ | √ | ? | No | 2,100<br>1,600 | 1,700<br>1,750 | 1,550<br>1,600 | Magnetic<br>To soften draw at 1,300-1,400 deg. F. | For general corrosion resistance where hardness is required. For cutlery, surgical instruments. |
| .... | .... | √ | √ | √ | √ | √ | √ | √ | 2,000<br>1,200 | .....<br>..... | 1,450<br>..... | Magnetic<br>Non-hardening | General corrosion-resistant material for fabrication. Chemical equipment, nitric acid towers. |
| 6,500<br>325 | 5,000<br>120 | √ | √ | √ | √ | √ | Plates | √ | 2,100<br>1,400 | .....<br>..... | .....<br>1,600 | Magnetic<br>Non-hardening | Special cases of corrosion resistance, temperatures up to 1,800 deg. F., for $SO_2$ and $SO_3$. |

cent, much less than for the fabricated forms.

| | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| .... | .... | √ | √ | √ | √ | √ | √ | √ | 2,000<br>1,200 | 1,500<br>1,600 | 1,450<br>1,500 | Magnetic<br>Non-hardening | General engineering purposes, low temperature turbine blading. |
| .... | .... | √ | √ | √ | √ | √ | √ | No | 2,000<br>1,200 | .....<br>..... | Quench | Non-magnetic<br>Non-hardening | Resistant but not immune to hydrochloric and sulphuric acids. Non-magnetic material for electrical parts. |

lades; has recently been superseded by the low carbon, 12 per cent chromium A-1 type.
with 10 to 12 per cent chromium is being used by turbine designers for higher temperatures than now prevailing.

| | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| .... | .... | √ | √ | √ | √ | √ | √ | √ | 2,100<br>1,400 | .....<br>..... | Air or Water quench 1,800 to 2,000 | Non-magnetic<br>Non-hardening | Chemical equipment; architectural, food, laundry, dyeing industries. |
| 9,000<br>450 | .... | √ | √ | √ | √ | √ | √ | √ | 2,100<br>1,400 | .....<br>..... | | Non-magnetic<br>Non hardening | General fabrication; riveted and welded structures for chemical equipment. |
| 10,850<br>540 | 5,500<br>125 | √ | √ | √ | √ | √ | √ | √ | 2,100<br>1,400 | ..... | | Non-magnetic<br>Non-hardening | General fabrication, resistant to sulphite solutions in paper processes. |
| 11,500<br>575 | 6,500<br>150 | √ | √ | √ | √ | √ | √ | Hollow Forged | 2,100<br>1,400 | ..... | | Non-magnetic<br>Non-hardening | For mixed acid conditions in paper, dye, and general chemical processes. Resistant but not immune to hydrochloric and sulphuric acids. |

vanadium and columbium are often added to retard or control carbide
Where some of these elements are present the materials may become slightly magnetic.

| | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| .... | .... | √ | √ | √ | √ | √ | √ | √ | 2,000<br>1,200 | ..... | Air or water quench 1,800 to 2,000 | Non-magnetic<br>Non-hardening | Resistant to salt water, cold dilute sulphuric acid. For oil refineries, naval equipment and general chemical uses. |
| 13,250<br>650 | 7,500<br>175 | √ | √ | √ | √ | √ | √ | Plates | ..... | ..... | | Non-magnetic<br>Non-hardening | Low chrome, for high temperature turbine blading. High chrome, for heat resistant material for carburizing boxes and furnace parts. |
| 15,250<br>750 | 9,250<br>225 | √ | ? | ? | ? | ? | ? | No | ..... | ..... | | Non-magnetic<br>Non-hardening | For temperatures up to 2,000 deg. F., for furnace parts, forgings, electrical equipment. Resistant but not immune to hydrochloric and sulphuric acids. |

Low chromium type only

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D-2 type with modifications is available in various forms. Most of this material is used for heat resistance.
F. D-3 type is obtainable only in restricted forms. Modifications of this type are obtainable in certain
ybdenum this material becomes immune to hydrochloric and sulphuric acids.

Note 2—The first figure is the ultimate stress obtained after one hour at temperature; the second figure is considered by the author to be a conservative design stress for use at these temperatures. These working stresses are based on experience and have been used satisfactorily. But they should not be confused with creep strengths.

# Selecting the Motor to Fit the Requirements—I

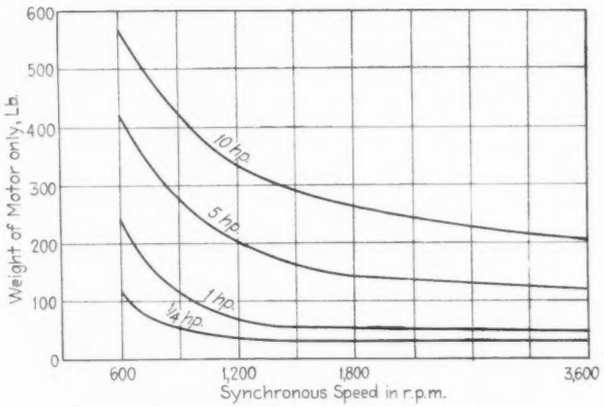
## A discussion of motor sizes, speeds, duty cycles and ventilation requirements

TWO general types of problems enter into the selection of a motor for a given application. One type has to do with the questions relating to motor characteristics while the other group of problems concerns itself primarily with mechanical features of the design of both the motor and the driven machine. Questions of the first type lead into the realm of electrical design and are answered for the product engineer by the specialists of the motor manufacturer. But the product engineer must furnish to the motor manufacturer all the information concerning the driven machine that may in any way affect the motor selection. In the final analysis, the responsibility for the selection of the proper motor belongs to the product designer. Hence he must not only be acquainted with the operating characteristics of the various types of motors but he must also have a fair idea of the possibilities and limitations in motor design.

Perhaps the first question for the designer to decide is the speed of the motor to be used. This usually is a simple question to answer. The speed of the shaft of the machine to be driven is usually fixed, at least within narrow limits. Also, motor speeds, particularly of a.c. motors, are quite definite. In the normal speed range, the full-load speed of a 60-cycle a.c. synchronous motor may be 865, 1,140, 1,725 or 3,440 r.p.m., depending upon the number of poles. Standard d.c. motors are manufactured for about the same speeds, though their speed permits of regulation. Ordinarily, if the driven shaft speed happens to be the same, or nearly so, as one of the synchronous motor speeds, a direct motor drive might be selected. But this is not always possible. For example, the running torque or the starting torque required of the driven machine may be high, making it advisable to use a high-speed motor with a drive through reduction gears.

Of the advantages in using high-speed motors, perhaps the most important one is size. As an example, the $\frac{1}{8}$-hp. 865-r.p.m. motor may use the same frame as a $\frac{1}{2}$-hp. 3,440-r.p.m. motor. Similarly, with the same size frame, an 865-r.p.m. motor develops $\frac{1}{3}$ hp. while a 3,440-r.p.m. motor has a $1\frac{1}{2}$-hp. rating.

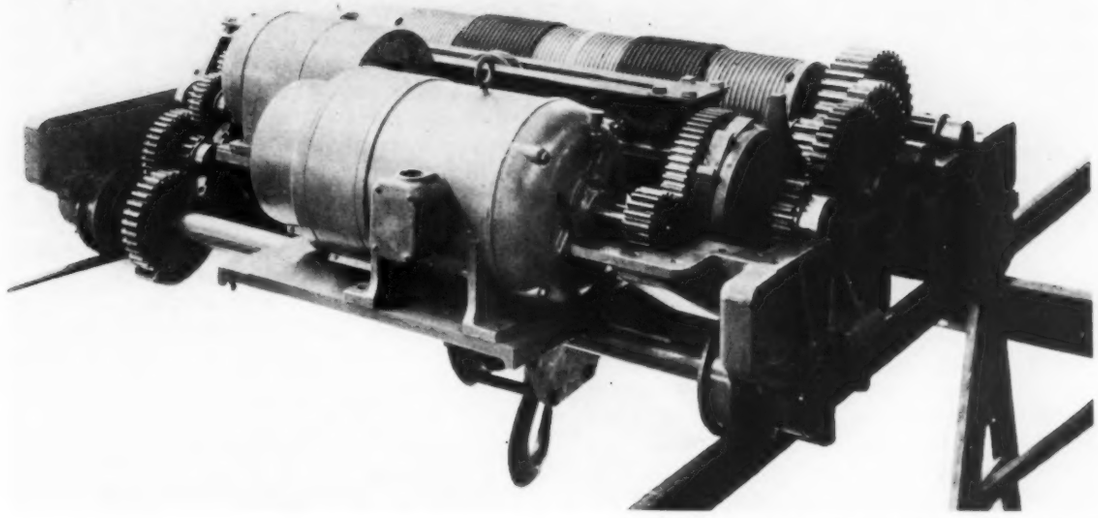
In the accompanying curves, synchronous speed in r.p.m. is plotted against motor size expressed in pounds weight. If motor volume had been used for the ordinates, the curves would have been practically identical. It will be observed that for the larger sizes of motors, the size of the motor decreases rapidly with increased speed between the range of 600 to 1,800 r.p.m. But for the smaller horsepower motors the variation in size for the different speeds is not nearly so great. Also, for all horsepowers, the size of the motor does not decrease appreciably beyond 1,800 r.p.m. For the fractional horsepower motors, the size of the motor at 1,800 r.p.m. and 3,600 r.p.m. is practically the same.




**Curves showing the relation between weight and synchronous speed of standard squirrel-cage motors ranging from ¼ to 10 hp. The greatest savings are made when increasing the motor speed from 600 to 1,800 r.p.m. Beyond this speed the weight saving is relatively little**

The saving in space by the use of a high-speed motor is somewhat offset by the space occupied by the reduction gears. There is also to be considered the power lost in the reduction gearing. But in the latest designs of "geared-head" motors, gear-motors, or motor-reducers, as they are variously called, the gears are extremely compact, adding only to the length of the motor. The usual object of the designer is to keep the diameters to a minimum; ordinarily space is available to take care of extra length. As to the power loss in the speed reducer, modern gearing can provide efficiencies better than 95 per cent for reduction ratios as high as 50 to 1. For the small fractional horsepowers and higher reduction ratios, the efficiency of the speed reducer is usually of no great importance.

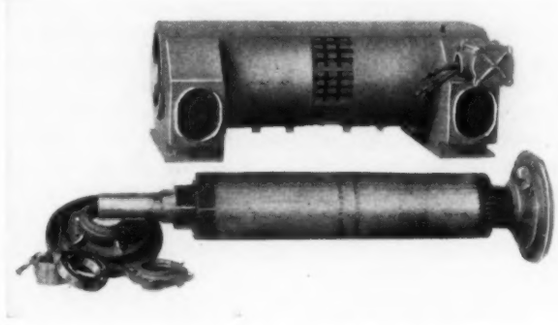
When a smaller diameter motor is desired, the length of the armature can be increased, thereby reducing the diameter. It is a relatively simple matter to lengthen a motor frame. Even standard motor designs resort to lengthening the armature in order to obtain increased horsepower without increasing the diameter. Thus the standard $\frac{1}{6}$-hp. and $\frac{1}{4}$-hp. motors are built to the same diameter. The difference in horsepower is obtained by the increase in the length of the armature. About the maximum that can be obtained practically is a range from $\frac{1}{3}$ to $\frac{1}{6}$ hp. for the same diameter rotor. Usually the increase in the length of the core is limited



Trolley of the Conco 5-ton crane with gear covers removed. In the foreground is a 15-hp., 1,665-r.p.m. Star hoist motor. Behind it to the left is a 1½-hp. rack motor. In addition to an electric disk brake on the hoist motor, an automatic mechanical brake is mounted on the intermediate shaft. All gears run in an oil bath and are mounted on Hyatt roller bearings

to 50 per cent longer than the standard length. As the length of the armature increases, the shaft deflection becomes a problem, and core losses increase as the ratio of length to diameter departs from the optimum. However, there has recently been placed on the market a line of motors ranging from 12 to 14-in. in diameter with a length of 40 in. for a 100-hp. motor.

With reference to the increased overall length of the motor when the rotor length is increased, a 33 per cent increase in the length of the armature increases the overall length of the motor only about 12 per cent. This is because the length of the armature represents only about ⅜ the overall length of the motor.



Louis Allis 1,800 r.p.m., 100-hp. motor, 40-in. overall length, 14-in. diameter. A separate blower is used for ventilation. This design has been developed for the mounting of grinder wheels, saws or cutterheads directly on the motor shaft extension

Roughly, a 12 per cent increase in motor length will result in about 30 per cent increase in horsepower.

Closely tied up with the question of speed is that of starting torque. Unfortunately it is often extremely difficult to calculate the torque required for starting a machine. It is not always a simple matter to calculate the moment of inertia of the various moving parts, and the coefficients of friction must usually be estimated. Where there is no past experience on which to base judgment, it may be better to hook up a temporary motor with meters in the circuit to determine the starting torque required of the machine. This, of course, cannot be done in every instance, but it is interesting to note that this is the method commonly adopted for determining the size of motor to be used on small devices such as vacuum sweepers, electric makers for the household, washing machines and similar appliances.

When a geared speed reducer is put between the motor and the driven machine, the inertia of the gearing increases the required starting torque. The energy that must be expended in bringing a wheel up to speed varies directly as its moment of inertia and as the square of the speed. There is, therefore, a considerable gain in having a small motor pinion with as large a speed reduction to the first gear as may be practical, both the pinion and the gear being made as small in diameter as possible. Through the use of high quality heat-treated helical type gears, the gear diameters can be reduced to a minimum with a corresponding decrease in inertia.

More particularly in small devices, the starting torque required to overcome inertia represents the bulk of the load. Under such conditions there is often an advantage in the use of V-belts when the required reduction ratio is within the limits of that type of drive. Because V-belt sheaves are considerably lighter than equivalent gears, the inertia to be overcome is correspondingly less. Also, the resiliency of the V-belt drive is a favorable factor.

Frequently it is necessary to start the motor with the full load on the machine. All of the early compressors in domestic refrigerators started in that manner. Unloader valves that keep the load off the machine until

it has reached full speed were then developed. These valves permitted the use of a considerably smaller motor because of the lower starting torque required. Practically all refrigerator compressors in use today are designed so that the motor starts under no load.

In all motor applications wherein the size of the motor is determined by the starting torque, and the machine starts under full load, much can be gained if the design can be changed to allow the motor to start with no load. This might be accomplished in a number of ways. As mentioned above, unloader valves can be used on almost all types of compressors. For other types of machines, centrifugal clutches of various types are available. Magnetic clutches can be used for some drives. All of these disconnecting devices can be applied so as to be operated automatically. In addition to the saving in the initial cost of the motor there is to be considered the advantage of smoother starting with less wear and tear on the equipment. In addition, the lower inrush current obtained with no-load starting may permit the elimination of starting resistance in the motor circuit.

It is not sufficient to consider only the starting torque and running torque. The real determining factor in selecting the size of motor is the "duty cycle." Frequent starts and stops, with only short intervals between, represent the most severe motor operating requirements and may demand an extra large motor. On the other hand, if the motor runs only for a short period, with long stops in between, a smaller motor than needed for continuous running might be used.

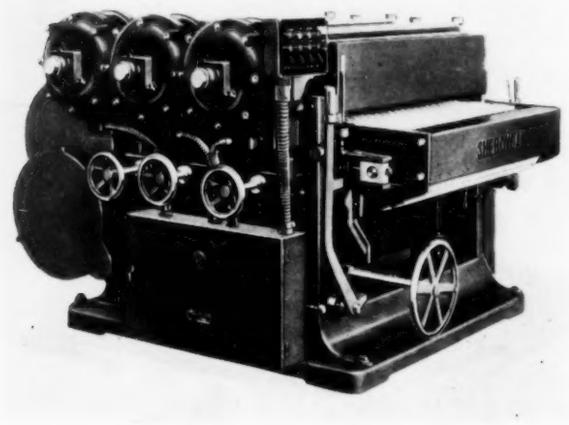
Finally, there is the question of overload or peak-load. Power presses, for example, have a short period in their cycle of operation in which the loads go extremely high. Such a condition must be considered in the selection of the motor, but it does not necessarily follow that a motor sufficiently large to carry such a peak-load continuously, must be used. All standard motors have a short time overload rating. For example, certain classes of standard motors can deliver a 100 per cent overload for a period of 15 seconds or a 50 per cent overload for a period of five minutes. Some of the larger sizes of motors can deliver a 50 per cent overload for a period as long as 15 minutes. When operating under such conditions, there must be sufficient time between overloads to allow the motor to return to normal running temperature.

In discussing the selection of motors, the motor designer of one of the leading manufacturers stated, "We must know everything that happens from the moment that the motor switch is closed until the motor has been shut down for the day." This not only refers to starting and stopping and the loads that might come on the motor, but also refers to all other operating conditions and all details concerning the motor mounting.

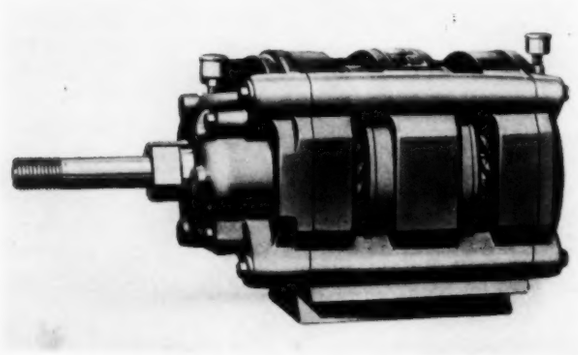
Perhaps the most important thing in a motor mounting is to provide sufficient ventilation to prevent the motor from burning out. The amount of ventilation required will vary considerably for the same horsepower motor in different frame sizes. The duty cycle of the motor is another controlling factor. Standard motors are designed for a temperature rise of 40 deg. C. Sometimes a 50 deg. C. temperature rise is permissible. But regardless of what the permissible temperature rise may be, it is evident that the temperature of the motor will increase until the rate of heat dissipation equals the rate at which heat is generated in the motor. The rate of heat generation in the motor is equal to the horsepower input less the horsepower output. Expressed in another way, the heat generated in the motor represents the power loss or inefficiency of the motor.

The size of the motor frame has, in itself, nothing to do with the amount of ventilation that will be required. This is strikingly illustrated by a comparison of a $\frac{1}{2}$-hp. 3,440-r.p.m. motor and a $\frac{1}{8}$-hp. 865-r.p.m. motor mounted in the same size frame. Assuming an efficiency of 50 per cent for the $\frac{1}{8}$-hp. motor and an efficiency of 70 per cent for the $\frac{1}{2}$-hp. motor, the $\frac{1}{2}$-hp. motor although delivering 400 per cent more power will require only 65 per cent more ventilation than the $\frac{1}{8}$-hp. motor. And both of these motors will deliver the same torque.

In standard motors the necessary ventilation of the motor is always provided but in applying the motor the product designer must see to it that there will be sufficiently free access of air to the motor. For built-in motors, the problem becomes more involved. A detailed discussion of this phase of the problem will be given in the next article of this series.



**In the Sheboygan Machine Company three drum sander each roll is direct connected to a Louis Allis 5-hp. 1,200-r.p.m. frame-type squirrel-cage motor. The motor frames are slab-sided as shown, making it possible to mount the sander rolls on 14-in. centers**



# How Photo-Electric Controls Are Applied—I

Operating characteristics and the relative limitations and advantages of the three types of photocells and their auxiliary equipment were given under the title "Photocells" in the August number. This is the first of a series of articles giving an analysis of mechanical movements controlled by photocells, with examples taken from actual applications

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BECAUSE the electrical power output of photocells is relatively small, if the operation controlled by the photocell is a mechanical one requiring considerable power, the photocell output cannot be used directly. Although some photocells can operate sensitive relays directly, other cells must have their output amplified through vacuum tube amplifiers and relay tubes. Even such amplified output is frequently insufficient and can be used only for the operation of relays controlling power circuits. In most applications of photo-electric machine control the relay-controlled power circuits are used merely to operate magnets, solenoids or motors which in turn actuate clutches, brakes, cams, pawls or similar mechanical devices. With such electro-mechanical control the power for the desired mechanical operation is usually supplied by the machine itself.

All photo-electric controls fall into one of two general classifications, each of which imposes restrictions on the selection of the type of cell and amplifier that may be used. Under the first classification come those applications wherein the photocell is required to furnish power continuously over an extended period of time. As an example, the photocell may be used to hold open the relay switch in a lighting circuit so long as there is sufficient illumination from other sources.

In the second general classification fall all those installations in which the change in the light intensity on the photocell, either momentary or of longer duration, generates an electric impulse which in itself may serve to do the required operation or may initiate a cycle of operations. Power for carrying through the cycle of operations is obtained when the photo control device causes engagement of the mechanism of the cyclic operation with some moving part of the driven machine.

Photo-electric safety controls on punch presses and similar machines illustrate the application of photocells wherein the cell output must be continuous over an extended period of time. The purpose of such safety control is to permit operation of the punch press only when the operator's hands are clear of a light beam directed across the front of the machine. An example of this type of safety control is illustrated diagrammatically in Fig. 2, with the various parts and units labelled.

Either of two methods may be used to prevent engagement of the dog clutch that drives the machine. A stop can be arranged to block the operation of the trip lever or some linkage can be devised to disconnect the trip lever from the clutch shifter. The latter method, illustrated in Fig. 2, has two distinct advantages. The operator cannot attempt to force the lever, and in addition, the operator will be penalized when an unsafe operation is attempted because it will be necessary to release the trip lever and repeat the operation.

As shown in the design in Fig. 2, the connection between the trip lever and the clutch lever is a bar hinged to the trip lever. When this bar is held against the stop by the pull of the solenoid, pushing down on the trip pedal will operate the clutch lever. If for any reason the solenoid fails to operate, the spring will pull the connecting bar away from the clutch lever. Thus, any failure in the photo-electric control, as well as an interruption of the light beam across the press, keeps the press inoperative.

With a light-weight hinged bar having a relatively small moment of inertia the energy required by the solenoid operated on a.c. is not more than 100 watts. The solenoid is connected through a contactor or

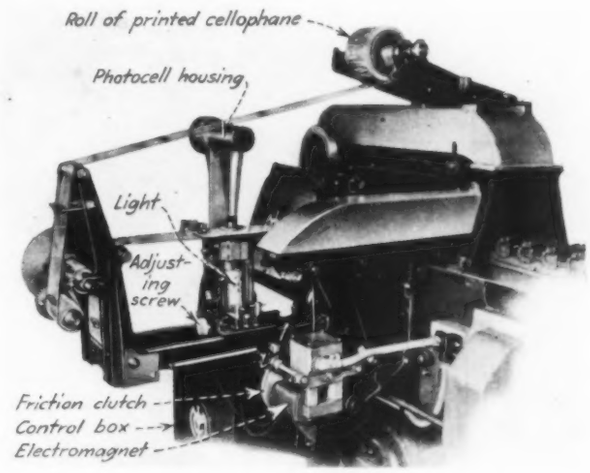


**Fig. 1—On this machine designed by the International Cigar Machine Company each printed band on the cellophane cigar wrapper is accurately positioned by photo-electric control.**

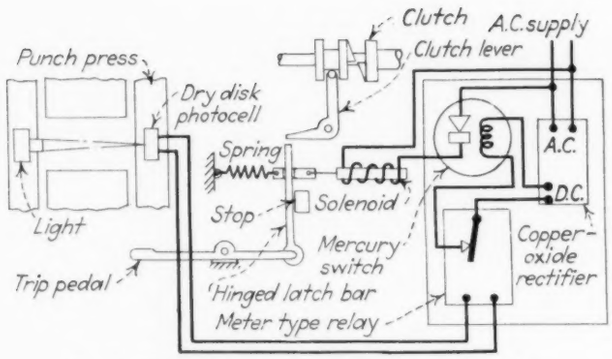


**Fig. 2—To operate the electromagnet on this safety device the dry disk voltaic cell requires a meter-type relay and a mercury switch**

mercury switch to the same a.c. power supply line that furnishes the power for driving the press. Mercury switches operated by d.c. require much less power than a.c. contactors, have fewer parts, and require little or no attention or adjustment. To provide a d.c. supply for operating the mercury switch, the full-wave copper-oxide rectifier, shown in Fig. 2, is inexpensive, has no moving parts, and does not involve the use of electronic tubes.

Because the speed of response is not an important factor in this installation, either of the three types of cells can be used. Dry-disk or voltaic cells have the advantage of operating a sensitive relay directly without the use of an expensive vacuum tube amplifier. In the meter-type sensitive relay used with the dry-disk cell a moving coil and pointer assembly is suspended in a strong magnetic field. Current from the photocell swings the pointer against an electrical contact closing the circuit to the operating coil of the mercury switch. Because of the light contact, the meter-type relay is not capable of controlling circuits carrying more than about $1\frac{1}{2}$ watts, which explains the necessity for the mercury switch.

A similar type of photocell control is used in controlling the sludge level in the settling tanks in sewage disposal plants. A motor-driven pump at a distant location removes the sludge when the accumulation has reached a predetermined level. Automatic control is obtained by means of a photocell and light source in separate water-tight housings submerged in the settling tank as shown in Fig. 3 and adjusted in height for the required sludge level.

Two factors limiting the selection of the photocell are dampness in the submerged cell housing and the distance between the tank and the pump motor. With either the selenium cell or the photo-emissive tube, current leakage introduced by dampness may cause unreliable operation. In addition, these cells both require short leads to the amplifiers making it necessary to install a long length of power wiring from the control panel to the motor. Dry disk cells are available with moisture-proof housings for use under water, and long leads between the cell and relays are permissible with this photocell. The low-voltage leads can be installed at much smaller expense than the power wiring which would be required with the control panel close to the photocell. This would be necessary if either the selenium cell or the photo-emissive tube were to be used.

In the control panel for the dry disk cell, as shown in Fig. 3, the contacts in a meter-type relay are held open by the current from the photocell when light is falling on the cell. When the light beam is interrupted a spring closes the contacts. To prevent unnecessary operation of the pump by momentary interruptions of the light beam a bimetallic time delay relay operates the mercury switch in the pump motor circuit only after the light on the photocell has been interrupted for about two minutes. The time delay relay also keeps the pump in operation for a short time after the sludge level has fallen below the photocell.

As in the photo-electric control on the punch press, a copper-oxide rectifier supplies the direct current power for operating the mercury switch and the time delay relay. Power for the complete operation of the control is obtained from circuits available at the motor, and no amplifying or rectifying tubes or transformers are needed.

Another operation maintained by the photocell is illustrated in a control for a container filling machine. Containers are placed on a scale pan beneath a gate arranged for two rates of discharge, a high rate of discharge used until the container is nearly filled and a trickle feed to bring the container up to the proper

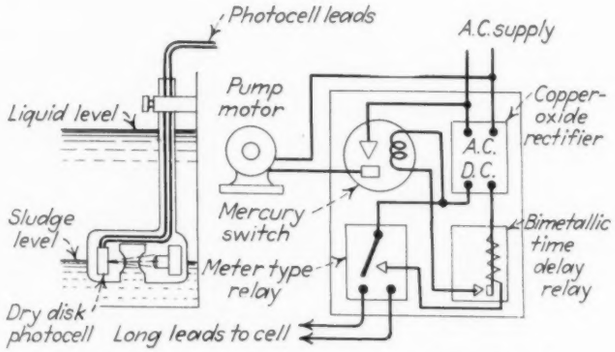


**Fig. 3—Long leads between the photocell and amplifier and the under-water location are factors entering the selection of the dry disk cell for this sewage settling tank**

weight. With manual operation, comparatively large variations in weight occur, but with photo-electric controls it is possible to increase the rate of filling and at the same time reduce the variations in weight.

A light beam directed on a photocell adjusted to the proper scale position is intercepted by the scale pointer, as shown in Fig. 4. The pointer also carries a colored transparent shield which reduces the light intensity on the cell before the pointer reaches its final position. Any of the various types of photocells can be used, although the complexity of the relay apparatus varies for each type. For a control using the dry-disk cell a double contact meter-type relay is operated by the cell. The zero position of the pointer arm on the sensitive relay is adjusted to make one contact with full illumination on the cell, but to take a position clear of both contacts when the filter is in front of the cell. No illumination on the cell causes the relay to make the other contact.

Mercury switches controlled by these contacts operate the feed valve solenoids allowing simultaneously

both full feed and trickle feed when full illumination is on the cell. When the filter on the scale arm intercepts the light beam the full feed is stopped; when the pointer obstructs the light both feeds are stopped. As in the other dry-disk cell controls direct current power to operate the mercury switches is supplied from an a.c. source by a copper-oxide rectifier. Feed valve solenoids are operated from the same a.c. supply.

For this control the sensitivity and the zero position of the meter-type relay must be adjusted for the opacity of the filter so that under restricted illumination the relay contact arm is clear of both contacts.

Selenium cells can be used for this application, but when selenium cells are used, the output of the photocell must be amplified to provide sufficient current to operate relays. Because of the relatively high power output of the selenium cell, a single amplifier tube operating on a.c. without a rectifier can be used as illustrated in Fig. 5. With an a.c. potential supplied to the photocell, the amplifier tube, being in itself a rectifier, will supply pulsating direct current in proportion to the resistance of the selenium cell, the cell resistance varying in proportion to the light intensity. The current from the amplifier passes through two sensitive magnetic relays connected in series. Spring tension on one relay is adjusted so that the contacts open when the relay current falls below 2 milliamperes; the other relay is adjusted to open when the current falls to 1 millampere. The two relays operate the gate valve solenoids in the same manner as described for the dry disk cell control.

One of two adjustments on the control panel varies the sensitivity of the selenium cell by regulating the a.c. voltage impressed across the cell; the other adjustment controls the value of the current delivered by the amplifier tube. Since selenium cells are subject to some aging, the sensitivity control may require adjusting each time the weighing machine is placed in operation. The amplifier adjustment is influenced by the characteristics of the particular amplifier tube and relays used. After being adjusted once, no further attention is required unless the tube or one of the relays is changed. The opacity of the red filter reduces the output of the amplifier about 75 per cent, and the amplifier adjustment is set so that the cell, when covered by the filter, causes only the 1 milliampere relay to operate, while with unobstructed illumination the cell operates both relays.

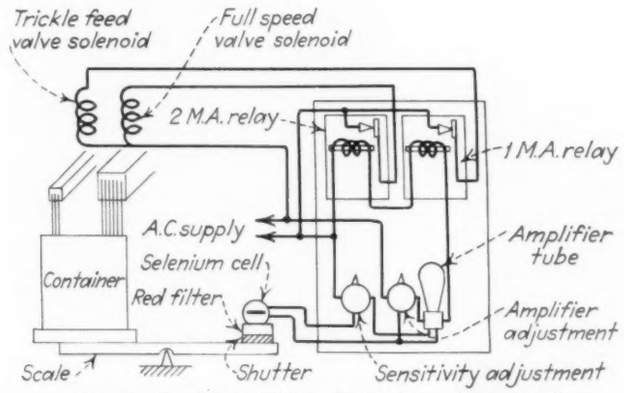


**Fig. 5—In this weighing device the selenium cell and amplifier perform the same function as the voltaic cell and meter-type relay shown in Fig. 4**

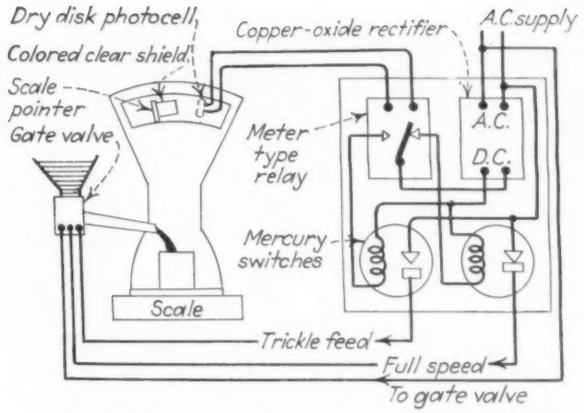


**Fig. 4—Photo-electric control successively changes full feed to trickle feed and then stops the trickle feed as the colored shield and scale pointer interrupts the light on the voltaic cell**

With reference to the possible use of photo-emissive cells for the above application, difficulties encountered in calibrating a photo-emissive cell and amplifier are often sources of trouble in application such as in this weighing device. Because of the high amplification involved in stepping up the phototube current, variations in supply voltage, tube characteristics, temperatures, current leakage, and other factors are much more serious sources of difficulty than with controls using the dry disk or selenium cell.

The photo-electric controls described above are for operations wherein the current from the photocell must be continuous over appreciable periods of time. In the second general classification, the photo-electric control supplies an impulse that initiates or performs a particular operation. Since only a momentary impulse is required from the cell, the speed of response usually imposes restrictions in the selection of the photocell and its amplifier and relays.

An example of high-speed initiation is illustrated in packaging or wrapping machinery for applying wrappers of printed cellophane to cigars, candy bars or packages of chewing gum. A roll of cellophane is printed with the designs properly spaced to provide the right length of wrapper for one package. In each cycle of operation these machines feed a length of paper, cut it off, and wrap it around a package. Thus the paper feed is intermittent, and the position of the printing on the wrapped article depends on the position of the printing in relation to the cutter. This relation must be corrected constantly when the machine is running, because of the effect of variations in temperature and humidity on the cellophane strip. When a manual control of an adjustable paper feed mechanism is used, the full time of an operator is required to keep the paper in register, and corrections can be made only after improperly wrapped packages have been delivered by the machine.

Several photo-electric methods to register the printing with the cutter will be described in the next number of *Product Engineering*. In all these devices the power needed to perform the operation is supplied by the machine, the photo-electric control serving only to engage the mechanism. In such applications the problem becomes one of ingenuity of mechanical design rather than photocell selection.

# COMMUNICATION and COMMENT

## Dry-Disk Type Photo-Electric Cells

*To the Editor:*

In the article on "Photocells," page 289 of *Product Engineering* for August, the discussion on the dry-disk type of photocell is limited to only one type of these cells. Some of the limitations and part of the information given does not apply to Weston developments in this art.

Copper or copper oxide is not used in the construction of the dry-disk type of photo-electric cell made by Weston Electrical Instrument Company under the trade name of "Photronic cell." Our research and development in connection with the various materials for dry-disk cells led us to abandon the copper-oxide type because of the errors mentioned in the article. The temperature coefficient of the photronic cell, as shown in Fig. 1, is relatively low so that for most practical purposes it can be neglected. Although the operating temperature of the cell is limited to about 120 deg. F., in order to avoid destroying the light-sensitive surface, under normal conditions of temperature and moisture the life of the cell is not affected by light intensities up to sunlight. Waterproof cells are obtainable where the cells must be used in a moist atmosphere.

If the photronic cell is connected to an instrument or relay with a resistance of about 25 ohms, the output of the cell has a linear relation to light intensity. The output of the cell, Fig. 2, varies with the resistance of the external load. For control purposes using an "on" and "off" cycle, an external resistance of 1,000 ohms or less may be used.

Although the dry-disk cell does not have the instantaneous response characteristics of the photo-emissive cell, the speed of response is rapid enough for many high-speed operations. For example, the Army Ordnance Department at Washington has used the photronic cell to time the travel of rifle bullets by operating a timing mechanism from the impulse obtained by the shadow of the bullet crossing the cell.

In using the dry-disk cell, no restriction is placed on the length of leads from the cell to the relay or instrument, provided the resistance is not higher than 10 per cent of the cell resistance. For example, in an installation in the Holland Tunnel between New Jersey and New York leads from the photocell are each 3,000 ft. in length. For industrial applications of the dry-disk cell it is therefore possible to mount the controlling mechanism on the wall or in some other convenient location, connecting the leads from the photocell to the control with conduit or other convenient wiring methods.

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*Weston Electrical Instrument Co.*

## Concerning Photo-Electric Control

*To the Editor:*

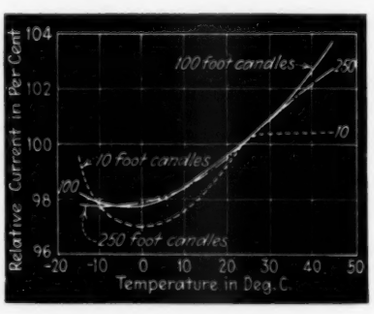
There is much admirable data contained in the article on photocells in the August *Product Engineering*. In judging the types of cells the summation is more fair than any that has appeared in some time. Some three or four years ago, when a number of the more popular photo-electric cells of today were first put on the market, their respective manufacturers were somewhat unrestrained in their advertising tactics, which was enhanced by the inherent romance of the possibilities of control with light.

Innumerable articles were published concerning things which had been accomplished and things that could be accomplished. These created the impression that photo-electric control was long since cut and dried and that the user needed only to find the money to purchase a cell in order to make himself a photo-electric expert. In addition, the nature of the early advertising was largely responsible for many impractical applications. Undoubtedly a large portion of those who purchased cells are now extremely dissatisfied with their investment and cannot be easily resold.

Constructive articles on the subject by presenting the whole truth can do much in the way of retracing steps and more or less repairing the damage done by the half-truths that have been told. No manufacturer is qualified to do this; but the editor who can take the pains to personally investigate the art and who can record his findings accurately can be of real service to his readers.

Little good can be accomplished by articles that tend to inspire the reader to harness the sun and distant stars, or by those articles which simply give broad summations of facts. The engineer, who is the ultimate user, could be better prepared by articles which analyze the installation, of which a photo-electric cell is but an insignificant part. Such analysis would also

**Fig. 1 — Variations in normal operating temperatures have relatively little effect on the output of photronic cells**



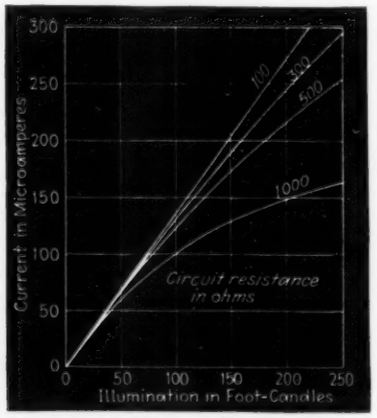




**Fig. 2—For circuits of low resistance the output of the photronic cell varies directly with the intensity of illumination**

clearly show that *it is the intelligence with which a cell is applied, not the type of cell, which determines the quality of the result.*—JOSEPH CLARK
*President, Clark Instrument Co.*

EDITOR'S NOTE—*We are in hearty agreement with the last sentence of Mr. Clark's letter. The editors of Product Engineering have been collecting data on numerous successful applications. The first of a series of articles based on the information gathered, appears in this number.*

## Chromium-Plated Taps

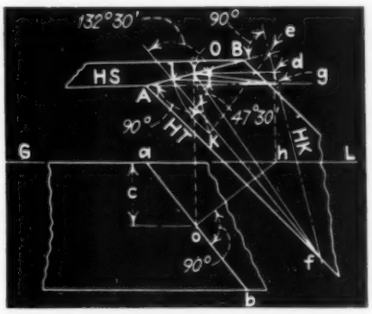
*To the Editor:*

In tapping a number of blind tapped holes in a molded bakelite part, spoilage was unduly high when standard taps were used. A spiral fluted tap made by annealing a twist drill, cutting a thread on the drill and rehardening, reduced spoilage to a negligible figure. When these taps were chromium plated, they completely solved the problem in regard to excessive wear and torn-out threads. It is a pity that such taps cannot be purchased complete, for there must be many firms which need them.—J. VINCENT.

## Hopper Problem

*To the Editor:*

The problem of finding the dihedral angles required for the hopper described in the May *Product Engineering* can be found by the ordinary descriptive geometry method. Using the top of the hopper as the horizontal reference plane gives the ground line *GL* as in the accompanying illustration. Through any point *O* on line *AB*, pass a plane *K* perpendicular to *AB*. This plane will cut out of the




**Method of applying ordinary descriptive geometry for finding the dihedral angle between hopper sheets**

two hopper surfaces the required angle.

The construction is as follows: Draw *oh* perpendicular to *ab*. Through its *H* piercing point *d*, and perpendicular to *AB* draw *HK*, the horizontal trace of the plane *K*. *HK* intersects *HS* at *g* and *HT* at *f*. Draw the angle *gof*, the top view of the required dihedral angle.

To obtain the true value of the angle, draw line *kO* perpendicular to *AB* and make its length equal to *c*. With *ck* as a radius, draw arc *kl* about *c* as a center. Angle *flg*, the required external angle, is 47 deg., 30 min.

—DEAN E. HOBART,
*Ann Arbor, Mich.*

# Materials That Merit Recognition

*To the Editor:*

It is to be regretted that for the article "Materials That Meet the Requirements," *Product Engineering* did not secure broader comment from a larger number of users of metals for operation at high temperatures. The information contained in the section "Metals for Service Under High Temperatures," appearing on page 219 of the June *Product Engineering*, is generally sound, although we consider that in some instances it is capable of misinterpretation by well-intentioned but comparatively uninformed readers.

For example, some of the paragraphs are not clear in that they do not sufficiently distinguish between rolled and cast products. Furthermore, they contain no mention of the high nickel-chromium alloys such as 66 per cent nickel-19 per cent chromium; and the 37 per cent nickel-17 per cent chromium alloys. The economy and dependability of these alloys has paved the way for most of the mechanical operations at extremely high temperatures such as encountered in mass heat-treating and kindred operations.

The design of mechanical handling equipment that operates at temperatures such as commonly used in hardening and carburizing is something more than an expansion or a development of previously existing ideas. Such design embodies a new type of engineering which uses the non-scaling and strength retention factors of the high chrome-nickel alloys. In addition, a thorough conception of the functioning of parts under stress at high temperatures and also under shocks occasioned by motion and load transfer is required. Provisions must be made for expansion and thermal stresses and ingenuity is required to protect parts from self-distortion which would result normally from stresses occasioned by unavoidable temperature differentials within unit parts.

This company, as founders and engineers of both the chrome-nickel and the nickel-chrome series, believes that proper recognition should have been given to the tremendous development of the nickel-chromium alloys that are serving industry in the really high temperature applications.

—G. C. MCCORMICK.
*Vice-President, General Alloys Co.*

In the main, we must agree with the above general criticism, particularly with reference to the dearth of information on the application of metals for service under extremely high temperatures. We must confess, however, that there were several other instances wherein we were forced to omit, because of lack of space, a proper mention of alloys for a specific service.

Concerning the alloys referred to by Mr. McCormick, interested readers will find a brief summary of these in the second installment of T. Holland Nelson's article, "The Chrome and Chrome-Nickel Iron Alloys, which appeared in the August *Product Engineering*. Furthermore, a more detailed consideration of the design problems involved in the application of metals for extremely high temperature service will appear in one of the forthcoming numbers.

As to the possibility of misinterpretation, as referred to in the first paragraph of Mr. McCormick's letter, it is almost impossible to avoid that risk in an article such as "Materials That Meet the Requirements," wherein the fundamental purpose was to give information concerning materials in use, and to recite the opinions and experiences of a large number of engineers. Also, it was necessary to include as wide a diversity as possible in the types of products and the service requirements that had to be met.

## Spring Design

*To the Editor:*

In his comments in the August *Product Engineering*, Frederick Franz suggests that there is no reason for running the risk of failure from high stresses of 90,000 lb. per sq.in. for springs made of small diameter wire subjected to infrequent stress cycles, yet these stresses are utilized in the

main spring of a wrist watch. It appears that if this stress is satisfactory for the wrist watch spring, it would be equally satisfactory in other applications. Springs in this class of service are subjected to working conditions but little different from static stresses, and the material under consideration is of high quality with an ultimate stress value of over 200,000 lb. per sq.in.

In the comparison of safe working stresses for compression and extension springs, certain practical considerations are involved. The weakest point in an extension spring is the point where the wire is bent to form the ends. In addition, extension springs wound with initial tension and having a high stress range per cycle are subjected to coil clash. These are sufficient reasons for reducing the allowable fiber stress under those permitted in compression springs.

It is true that if both force and dimensions are specified, a spring may be produced that is within the dimensional limits but outside the load-deflection limits. But complete dimensions of the spring are required in order that it will fit into the space provided, and it is also necessary that the force be of such magnitude as to properly support the load, so that both factors must be specified. Any error which the designer may have made in his calculations will thus be discovered by the spring manufacturer when both the dimensions and the load are specified within limits.

Exception is taken to the statement that the best proportioned helical extension or compression spring has an index of about nine. With other conditions equal a spring with an index of two will not give the same satisfactory service as a spring with an index of nine.

—W. M. GRIFFITH
*Executive Engineer*
*Atlas Imperial Diesel Engine Co.*

## Gear Problem

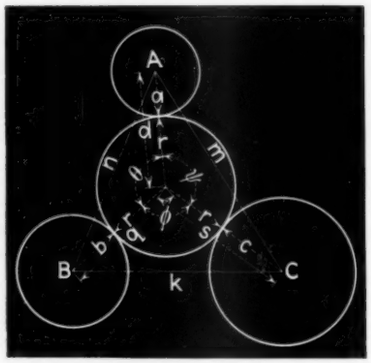
In the August number C. A. Wiken submitted a design problem involving the radius and location of a common drive or mongrel gear for a multiple spindle drilling machine. In each of the two solutions given here a general formula for the gear radius is developed by the use of the cosine theorem, although the two formulas are of different form. The use of these formulas necessarily involves long drawn calculations to obtain the answers correct to four decimal places, probably accounting for the differences between the two sets of results.

*To the Editor:*

This is a tedious problem to solve and care must be taken in using the formula to substitute the values exactly as they appear in the figure. Since the results are to be accurate to four decimal places, it is necessary to use eight place logarithm tables in all computations. The radius of the mongrel gear can be determined by substituting the given values in the general formula, given herewith as Equation (3). This equation is developed as shown in the accompanying table, making use of the cosine theorem. Substituting in Equation (3) the values given with the problem and solving for $r$ gives two values, the larger value being the radius of the circle making external contact with all three circles, and the smaller value the radius of the circle making internal contact. The smaller value of $r$, as required for this problem, is 1.3443 in.

Again using the cosine relationship, Equation (4), the angle $ABO$ is found to be 20 deg., 50 min., 2 sec. With this angle known, the required dimensions $x$ and $y$ are readily determined. Carried out to the required four decimal places, these values are: $x = 1.5246$ in., $y = 2.2818$ in.

—W. W. JOHNSON
*Cleveland, Ohio*




$\cos\theta = \cos(\phi+\psi) = \cos\phi\cos\psi - \sin\phi\sin\psi$ or,
$\cos\theta - \cos\phi\cos\psi = -\sin\phi\sin\psi$
Squaring both sides and simplifying,
$\cos^2\theta + \cos^2\phi + \cos^2\psi - 2\cos\theta\cos\phi\cos\psi = 1$ (1)

$$\cos\theta = \frac{p^2+q^2-n^2}{2pq} = 1 + \frac{(p-q)^2-n^2}{2pq} = 1 + \frac{(a-b)^2-n^2}{2pq}$$

$$= 1 + \frac{J}{2pq} \text{ where } J = (a-b)^2 - n^2$$

Similarly,

$$\cos\phi = 1 + \frac{K}{2qs} \text{ where } K = (b-c)^2 - K^2$$

$$\cos\psi = 1 + \frac{L}{sp} \text{ where } L = (c-a)^2 - m^2$$

Substituting these values in equation (1)

$$J^2s^2 + K^2p^2 + L^2q^2 - 2JKsp - 2KLpq - 2JLqs - JKL = 0 \quad (2)$$

Substituting $p = r+a, q = r+b, s = r+c$ in equation (2)

$$Tr^2 + Ur + V = 0 \text{ where} \quad (3)$$

$$T = \left[J^2 + K^2 + L^2 - 2JK - 2KL - 2JL\right]$$

$$U = \left[J^2c + K^2a + L^2b - JK(a+c) - KL(a+b) - JL(b+c)\right]$$

$$V = \left[J^2c^2 + K^2a^2 + L^2b^2 - 2JKca - 2KLab - 2JLbc - JKL\right]$$

$$\cos\tfrac{1}{2}\angle ABO = \sqrt{\frac{h(h-p)}{qn}} \text{ where } h = \frac{p+q+n}{2} \quad (4)$$

**Equations for the gear radius are developed from the cosine theorem in Mr. Johnson's solution**

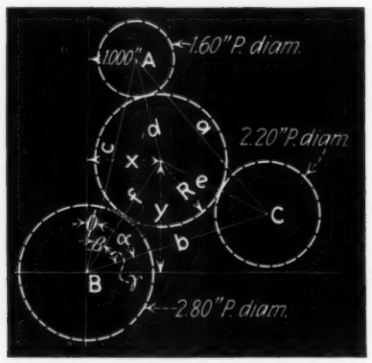
*To the Editor:*

There appears to be no shortcut method to the problem presented. The following analytical method, although laborious, may be carried out to any degree of accuracy. Using the notation as given on the accompanying diagram the cosine theorem provides a general solution in the form of the equation given with the accompanying figure. Substituting values for $a$, $b$, $c$, $d$, $e$ and $f$, as given in the problem, the equation becomes,

$.741256R^4 + 3.692438R^3 + 2.465708R^2 - 6.667401R - 6.882751 = 0$

Solving by Horner's method, the radius $R$ is 1.34425 in. Applying the cosine theorem to the triangle with sides $c$, $d$ and $f$, *the* angles $\beta$ and $\phi$ are determined in terms of the given quantities. The required distances $x$ and $y$ are then readily found from the values determined for the angles $\phi$ and $\beta$ and the distance $f$. From the data given with the problem, these distances are as follows: $x = 1.5240$ in., $y = 2.2822$ in.

—MANIOUS GOTTLIEB
*Philadelphia, Pa.*




$$\sqrt{\frac{\frac{a+b+c}{2}\left(\frac{a+b+c}{2}-a\right)}{bc}} = \sqrt{\frac{\frac{f+d+c}{2}\left(\frac{f+d+c}{2}-d\right)}{fc}} \sqrt{\frac{\frac{b+f+e}{2}\left(\frac{b+f+e}{2}-e\right)}{bf}}$$

$$- \sqrt{\frac{\frac{f+d+c}{2}\left(\frac{f+d+c}{2}-f\right)}{fc}} \sqrt{\frac{\frac{b+e+f}{2}\left(\frac{b+e+f}{2}-b\right)}{bf}} \quad (1)$$

$$\cos\tfrac{1}{2}\beta = \sqrt{\frac{\frac{f+d+c}{2}\left(\frac{f+d+c}{2}-d\right)}{fc}} \quad (2)$$

**General equations for finding the gear radius $R$ and its location as given by Mr. Gottlieb**

## Technicalities in Nomenclature

*To the Editor:*

A comment appearing on p. 2 of the August *Product Engineering*, criticizing the terminology used in the article, "Graphical Analysis on Cam Roller Mountings," is correct from a strictly technical viewpoint, but the use or

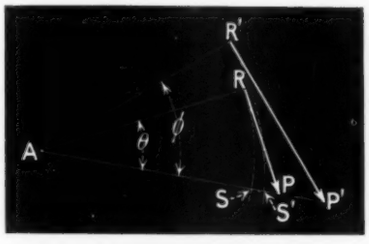
necessity of extreme technicalities in a subject for practical engineers is questionable. This is especially so when a strict observance of definitions involves long explanations which may obscure the point and have no effect upon the conclusion.

In the study of mechanics it is necessary that the students understand the correct terminology and proceed step by step through force, time and distance; in practical engineering the designer is more concerned with results than with intermediate factors, and subconsciously leaps from applied power to its effectiveness. Forces in mechanism mean little to the designer who thinks in terms of power rather than the separate elements of force, distance and time. Machinery's Handbook says:

"The term power is frequently used by writers on mechanics to designate force. In connection with the so-called 'mechanical power'—the lever, wheel and axle, wedge, screw, etc. — it is usual to speak of the applied force as the power."

The article on cam roller mountings, was written with the twofold purpose of presenting a subject given little consideration in texts and handbooks, and to offer a simple, visual solution eliminating time-consuming construction, measurements and mathematics. The purpose was to provide a short-cut on a practical subject in which only basic equations were given.

In the second equation on p. 251 of the July *Product Engineering*, $AP$ and $AR$ are each a force. They do not represent power, which is force times distance divided by time. But time and distance can be said to cancel out, since they are the same for $AR$ in all constructions that might be made. Cancelling out time and distance leaves the applied force, which is in proportion to the power. The same reasoning applies to the third equation. In the accompanying diagram the arcs $RS$ and $R'S'$ may or may not coincide in the two layouts having angles $\theta$ and $\phi$. Force components $AP$ and $AP'$ are proportional to the secant of the angles $\theta$ and $\phi$. The relative efficiency $AP/AR$ as compared to $AP'/AR'$ is evident. Since $SP$ and $S'P'$ are proportional to $AP$ and $AP'$ or proportional to sec$\theta$ and sec$\phi$ the relative efficiency is dependent upon the angle subtended between the direction of thrust and direction of motion.




**Although the vectors indicate forces, they can be considered as representing power if the time and distance factors are the same for all constructions**

In a preliminary to the explanation of the practical visual short-cut given in the article it seemed unnecessary to go through this analysis.

—E. Chat. Shanks
*Milwaukee, Wis.*

## Maximum Shear Stresses

*To the Editor:*

In the articles on photo-elastic analysis of stresses, by A. G. Solakian in recent numbers of *Product Engineering*, it is apparent that maximum shear stresses can be readily determined with a polariscope. In applying the results of photo-electric analysis to machine design problems, however, there are some factors that are difficult to reconcile. For example, in the November *Scientific American* an article on photo-electric stress analysis states that:

"It is a fortunate coincidence that this maximum shear stress, which is very easily obtained by the photo-elastic method, is the only stress of practical significance in ductile materials such as mild steel, because experiments show that such materials fail when the maximum shear stress reaches a critical value. Thus in machine design the strength of parts is based upon the maximum shear stress and not upon the principal stresses."

For mild steel, a ductile material, the average ultimate tensile strength is 60,000 lb. per sq.in., so that a bar of mild steel in tension would have a maximum diagonal shear stress at 45 deg. of one-half of this value, or 30,000 lb. per sq.in. According to the above quotation, the mild steel bar should fail by this maximum shear stress, but the ultimate shear stress of mild steel as determined by tests is approximately 50,000 lb. per sq.in. Is it possible that the lower value of shear strength for mild steel as compared to the tensile strength is caused by the stress concentration at the shearing edge? The failure of the crystalline structure in ductile material should occur at the same critical value of stress, whether in tension or in shear.—P. N. Bland
*Vancouver, B. C., Canada*

# *DEFLECTIONS*

REFRIGERATOR PROVIDES HEAT

At the Ancoats Hospital, Manchester, advantage is being taken of the fact, usually overlooked, that a refrigerator machine supplies heat in convenient form for utilization. A methyl chloride apparatus is operated to maintain a room temperature in one part of the hospital at about freezing point, but the condenser of the machine, instead of being cooled in the open air, is inclosed in a duct through which a fan draught is provided, the effluent warm air being delivered to the linen cupboards, for drying purposes. Cost of heating thus is negligible. Cost of refrigeration and heating is proportionately lower than if both functions of the cooling machine were not utilized.

PERPETUAL MOTION FROM M.I.T.

In a press release from the News Service of the Massachusetts Institute of Technology comes the important announcement of a servo-mechanism applying a novel torque amplifier. Quoting the press release "This motor through the torque amplifier, drives the slit by means of a screw and nut, the torque amplifier serving merely to increase the horsepower effect of the motor." We may be wrong, but it sounds like getting something for nothing.

DO ALL ENGINEERS ADMIT IT?

Seen in the engineering department of a well-known manufacturer of motor-control equipment, a sign reading:

"To err is human but not engineering"

Apparently, even some engineers admit that they're not human.

FOR NIGHT FLYING

New activities bring demands for new products. Night flying brought up the question of determining the "height of the ceiling" at night, the information then to be radioed to pilots in the air. The obvious answer was to throw a spot of light on a cloud and determine its height by observation and trigonometrical calculation. This accounts for an order for 25 ceiling projectors rated at 650,000 candlepower each, recently received by the Westinghouse Electric & Manufacturing Company. The projectors will be installed in various Weather Bureau Stations at airports throughout the United States and its possessions.

# · NEWS ·

## A Plan for Extending Engineering Employment

A recent statement by W. A. Shoudy, general chairman of the Professional Engineers' Committee on Unemployment, New York, contains so much of vital importance to designing engineers that we reproduce it herewith.

"Each new industry is a product of the engineer. He has not always been the first to conceive it, but he has perfected the mechanical details and in the last generation has formulated the principles of successful business management. Except during this depression, each threat of technological unemployment has been met with a new industry which absorbed both the old employee and the increasing supply of wage earners. In 1923 the American Management Association was warned of increasing unemployment, but even the engineer had been inoculated with the virus of speculation and this warning went unheeded. Through the past generation, the demand by these new industries for engineers has been so great that few have hunted for jobs; they have waited until a new one has turned up and this period of waiting has seldom been long.

"Because of this demand, the engineer has lost the skill in job-making for himself—the job has sought him.

"When the engineer is working at his profession, he is filling a place that cannot be filled by another type of mind. He is so often identified by his technical education that his other essential qualities are overlooked. He must have an analytical mind, an ability to recognize facts and to reason from facts only, intellectual honesty, courage to fight for his conclusions, and be able to 'do for one dollar what any one can do for two.' The individual engineer may have these qualities in varying degree but without a reasonably high rating he fails to be recognized as an engineer. His employment always results in a financial return to the employer, either as an increase in net earnings or as a prevention of waste.

"This Committee has, from its organization, continuously canvassed industry to locate unfilled engineering openings and proposes to carry on this activity. The members are convinced that this effort is not enough, but that entirely new engineering opportunities must be discovered, or re-employment among engineers will lag as business recovers. There can be little dispute about the statement that every existing industry can now employ additional engineers with a direct profit from such employment. The Committee proposes to study each industry to discover unrecognized openings for the engineer. These openings must:

1. Not interfere, for the present at least, with anyone now employed.
2. Be positions bringing an attractive financial return to the employer.
3. Be positions which only the engineer can fill.

"There are two major obstacles in the discovery of suitable openings of that kind:

1. The employer must be convinced of his need by showing him the large losses which he is now experiencing through lack of engineering assistance.
2. The unemployed engineer is discouraged by continuous and unsuccessful job hunting. Only a few understand the technique of job-finding and these few have at least continued to find means for support. He must be shown the opportunities for usefulness and taught the technique of securing this opportunity for himself.

"The proposed plan is as follows:

1. The selection of one engineer from each industry, or branch of the profession, to organize a committee within the industry.
2. This committee to study the industry to discover opportunities for engineering usefulness.
3. To list these opportunities and specify the qualifications of the engineer to fill these positions.
4. To send this list and specification to the P.E.C.U.
5. To prepare written arguments to employers showing the advantages of employing engineers for such positions.

"The P.E.C.U. will then notify properly qualified engineers that such opportunities are available in this industry and advise the engineer to:

1. Study the industry.
2. Study himself.
3. Prepare in writing a definite statement of the service he can render the proposed employer.
4. With the assistance of the P.E.C.U. the employing individuals in the industry will be found for the engineer but he must secure the position through his own efforts in presenting logical reasons for such employment.

"In the meantime the P.E.C.U. will have circulated the prospective employers in an effort to convince them of the need of employing engineers. They will further help the engineer by advice and in studying the industry but will insist that each man secure his place by his own efforts.

"One class of unemployed cannot be taken care of through this plan, viz.: Those who, through lack of broad education or too close specialization, find themselves with experience no longer required by industry. These men are a problem in rehabilitation that must be undertaken by a special committee with the most expert guidance and do not at this time form a part of this discussion.

"No attempt will be made to organize all industries at one time. Only as rapidly as suitable personnel for the committee has been secured, will the studies be started. Further, no attempt will be made to extend this work nation-wide, until at least one industry is organized. This work must be done thoroughly, bearing in mind that a few mistakes in placing unqualified engineers in new activities will do untold harm. However, the local sections of the national engineering societies will be kept informed of progress."

## Invisible Stars Detected by Photocells

A further refinement in astronomical instruments that will have far-reaching effects has been announced by Hollis Baird, chief engineer of Shortwave and Television Laboratory. In this new development, a photoelectric cell is used for comparing and classifying stars according to their brilliancy. Quoting Mr. Baird:

"A star must be classified according to its brilliancy or light magnitude. The human eye being variable in its effects cannot give an accurate reading of the relative light values of two weak distant starts. The photocell has provided the means that will actually measure the light of the stars without the variable link of human mechanism.

"Two tiny plates are used, one fixed and the other very delicately balanced. These are connected electrically so that a microscopic amount of electric current will cause them to repel each other. One being fixed, the other will move away from it, the distance it moves being determined by the amount of current received. This movement is marked off in degrees. So tiny is the actual movement that the degrees

must be read through a microscope. This complete device is called a photometer.

"The photocell registers accurately and by comparative readings stars can be classified. In addition, the cell is so much more sensitive than the human eye that it will detect stars invisible even through the most powerful telescope.

"Curiously enough, the man responsible for the equation which made the photocell possible is a man who by other equations has played hob with man's concept of space, being none other than the great Einstein. He is best known because of public interest in his theories on the abstract, yet his more practical contribution goes generally unrecognized.

"In 1905 Einstein presented the technical world with his photo-electric equation, which states that a given amount of light will cause a photocell to give forth a given amount of electrons. He received the Nobel prize for this work which conservative textbooks consider as important to the modern quantum theory as Newton's second law of motion is to mechanics. From this has sprung the photocell and its great contribution to the electronic art, with the astronomical electroscope one of its major contributions."

## Patent Decision on Lubricating Systems Reversed

A decision rendered by the Second Circuit Court of Appeals holds that the Alemite centralized lubricating system as manufactured by the Alemite Corporation of Chicago, is not an infringement of ten patents owned by the Auto Research Corporation of Delaware, and under which the Bijur Lubricating Corporation of New York is licensed. This decision is a reversal of a previous decision of the lower court. It permits the Alemite Corporation to enter the field of centralized lubrication for automobiles and industrial machinery, a field from which the Alemite Corporation would have been excluded if the decision of the lower court had been sustained.

## A.S.A. Takes Over Several Standard Bureau Activities

Joint announcement of the transfer of certain Bureau of Standards activities to the American Standards Association has been made by Secretary of Commerce Roper and Howard Coonley, president of A.S.A. These activities include those centered in the Divisions of Trade Standards, Specifications, Simplified Practice, Safety Codes, and Building and Plumbing Code Sections of the Building and Housing Division.

Drastic curtailment of Department of Commerce appropriations made it necessary to dispense with many activities that have come to be looked upon as fixtures by business and industry. Recognizing the importance of continuing as many as possible of these activities, government officials sought private organizations willing and able to carry them on.

In this particular instance the result of the combination of two standardizing units into one is looked upon by those concerned as a constructive step, especially in view of the acceleration of standards work likely to accompany NRA code adoptions.

Efforts are being made to keep Bureau experts connected with the work with which they are familiar by transferring them from the Bureau to the American Standards Association.

# *MEETINGS*

**National Metals Exposition**—Oct. 2-6, Detroit, Mich. W. H. Eisenman, secretary, 7016 Euclid Ave., Cleveland, Ohio.

**American Welding Society** — Annual fall meeting, Oct. 2-6 Book-Cadillac Hotel, Detroit, Mich. Miss M. M. Kelly, secretary, 29 West 39th St., New York, N. Y.

**Association of Iron and Steel Electrical Engineers**—Twenty-ninth convention, Oct. 17-19, William Penn Hotel, Pittsburgh,. Pa. John F. Kelly, managing director, Empire Bldg., Pittsburgh, Pa.

# *EXHIBITIONS*

**National Electrical Exposition** — Madison Square Garden, New York, N. Y., Sept. 20-30. Ralph Neumuller, managing director, 480 Lexington Ave., New York, N. Y.

**Second American Exposition of Brewing Machinery, Materials and Products**—Auditorium Hotel, Chicago, Ill., Sept. 23-Oct. 1. J. R. Nicholson, 55 W. 42d St., New York, N. Y., secretary, Master Brewers Asociation.

**National Metals Congress**—Oct. 2-6, Detroit, Mich. W. H. Eisenman, secretary, 7016 Euclid Ave., Cleveland, Ohio.

**Iron & Steel Exposition** — Oct. 17-19, William Penn Hotel, Pittsburgh, Pa.. John F. Kelly, managing director, Empire Bldg., Pittsburgh, Pa.

## Corrosion-Resistant Cast Iron

An austenitic cast iron developed in England is said to possess the corrosion resistance of the brasses and bronzes, but to have wear resistance equal to ordinary cast iron. This new cast iron was developed for applications requiring corrosion resistance and good wearing properties, particularly in sea water, acids and alkalis. It is reported that when immersed in aerated sea water for more than 2,000 hr. along with specimens of Admiralty metal and a good grade of ordinary cast iron, specimens of this austenitic cast iron show corrosion resistance somewhat better than the Admiralty metal and ten times that of ordinary cast iron. The new cast iron is also claimed to be less subject to wear from both erosion and cavitation than the softer Admiralty metal, brasses and bronzes.

Microscopic examination of the new alloy is said to show that this austenitic cast iron is more homogeneous, and that nearly all of the carbon present is in the form of a solid solution; the matrix is said to be more resistant to corrosion because of alloy additions. Because this new cast iron costs less than Admiralty metal, is harder and equally as good or better in its resistance to corrosion, it is expected to supplant Admiralty metal in some applications.

## Tungsten Carbide for Valve Stems and Seats

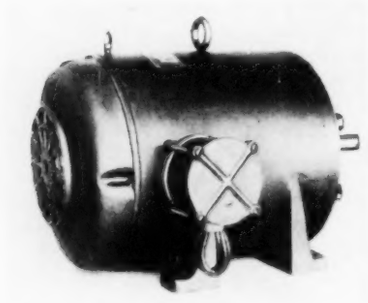
Because the valves stems and seats used in hydraulic presses, pumps and rams are subjected to high abrasion and great pressure, one manufacturer uses tungsten carbide for protecting those portions of the valve and stem that are subjected to the most severe abrasion and pressure.

The tungsten carbide is applied to the valve stems by brazing a small piece or blank to the section of the shank which is subjected to most wear. In the case of valve stems tungsten carbide replaces steel at the point-end only. For valve seats, a ring of tungsten carbide is used at the section where the valve contacts the seat. Sufficient thickness of metal is provided to allow repeated regrinding.

# *NEW MATERIALS and PARTS*

### Ideal Totally-Inclosed Fan-Cooled Motor

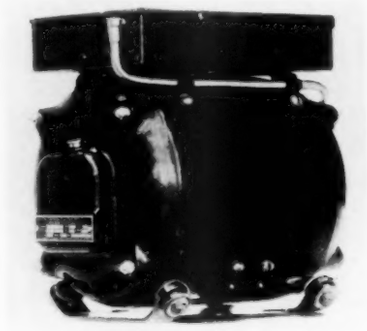
Is a fully inclosed motor of the same size as the ordinary open-type motor of the same rating. Claimed that protection from explosive dust, abrasive dust, corrosive gases or fumes, steam or moisture adapt this motor for use in steel mills, metal-working plants, foundries, packing plants, grain elevators, flour mills, chemical plants, laundries. Construction uses two shells, a dust-tight inner shell inclosing the windings and rotor, without running seals. Air is forced between the



inner shell and the outer one by a fan mounted between the shells. Available in all sizes from 1 to 200 hp. as squirrel-cage induction motors to meet all classifications of torque and inrush requirements, and with 1, 2, 3 or 4 speeds. Available also as across-the-line start motors up to 200 hp., and can be supplied for power factor correction. Ideal Electric & Mfg. Co., Mansfield, Ohio.

### Emerson Capacitor Motors

For refrigerators, oil burners, pumps, and compressors. Motors are designed to provide performance characteristics of a repulsion-induction motor without the use of brushes and commutators, providing greater quietness and less complication. Overload capacity in excess of 100 per cent is said to be provided in this design. Elimination of brushes and commutators and closer tolerances in air gaps is claimed to reduce electrical and mechanical noises and to avoid radio interference. Spring and rubber-base mountings are designed to absorb vibration. Large oil reservoirs with wool-packed bearing assembly is used. Connections are provided for changing direction of rotation, and frames for a.c. and d.c. are interchangeable. Condenser and terminal box mounted on top of motor, but can be supplied separately. Available in sizes from $\frac{1}{8}$



to $\frac{3}{4}$ hp., with or without resilient mounting. Emerson Electric Mfg. Co., 2018 Washington Ave., St. Louis, Mo.

### Tagliabue Dial Thermometer

With a moving element or bellows for use with vapor tension bulbs between minus 40 and plus 335 deg. F. Supplied in a black bakelite case with a black dial, 5-in. in diameter. Moving element consisting of a bronze bellows transmits motion to the pointer by a connecting mechanism said to be vibration-proof. Available with either brass or steel bulbs supplied with 5 or 10 ft. of spiral brass tubing in six

temperature ranges. Bulbs are $\frac{1}{2}$-in. in diameter, $5\frac{1}{4}$ to $7\frac{1}{4}$ in. long, fitted with a $\frac{1}{2}$-in. brass or steel union connection. C. J. Tagliabue Mfg. Co., Park & Nostrand Aves., Brooklyn, N. Y.

### Automatic Timer

For automatic and remote control can be arranged to operate a signal or terminate a process at the end of a predetermined period. The timing element is a telechron motor, self-starting and synchronous, requiring no winding, regulating or oiling. Resetting is automatic when the control circuit is de-energized. Two scales are provided, one with a range three times that of the other, and the timing period is adjustable over the entire range. Timer is set in operation by energizing a circuit in the

timer by pushbutton or switch. The operating pointer is then driven up the time scale and operates the timer contacts when it reaches the point at which the contacts are set. It is said that repeated operations of the timer with the adjustable contact set at the same point on the scale will show a variation of not more than 1 per cent. Available in three time ranges, and for 60, 50 or 25 cycles, 115 or 230 volts. General Electric Co., Schenectady, N. Y.

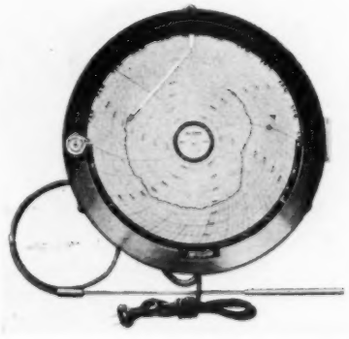
### DeVry Photoelectric Relay

For burglar alarms, fire alarms, counting and sorting material, opening and closing doors, controlling lighting systems. Consists of two similar units, the relay device, and a light source. At the base of the relay unit, a variable resistance operated by a knob at the back adjusts for various degrees of sensitivity. Photocell used is highly sensitive to infra-red light. Sockets in the base are for the relay operating contacts for closed or for open circuit. Primary relay rated to handle a $\frac{1}{2}$ amp. at 110 volts. An at-

tachment plug and 7-ft. cord are included. The light source has a 32 candle-power 6-volt headlight bulb. Cases for both units are supplied with a durable black crackle finish, and are 6½x5½x4½ in. Herman A. DeVry, Inc., 1111 Center St., Chicago, Ill.

## Recording Thermometer and Pressure Gage

Available in a complete line with 8-in. and 12-in. circular charts in indicating and recording types for temperatures from −40 deg. F. to 1,200 deg. F., and indicating or recording pressure and vacuum gages for ranges from 10 in. of water up to 5,000 lb. per sq.in. All types are offered in one, two or three pen models. For thermometers the mercury-filled helix is built of stainless steel said to provide for 100 per cent overload. The gas and vapor type helices are made of heat-treated phosphor bronze. Thermometers are designed to be located at distances up to 200 ft. In the pressure and vacuum gages, for ranges 10



in. of water to 30 lb. per sq.in., a flexible metal diaphragm is used; for ranges of 30 to 200 lb. per sq.in., pressure element is a spring-opposed bellows; over 200 lb. per sq.in., a helix is used. The chart is driven by an electric clock provided with a toggle switch for starting and stopping. Hand-wound clocks are supplied where a.c. current is not available. Cases are of die-cast aluminum and are designed to be interchangeable for back or bottom connection or for flush mounting. A built-in lock is combined with the door handle. Chart is pressed on the chart hub; no chart knob is used. Automatic pen release lifts the pen from the chart when the door is opened. Available with a large variety of bulbs for temperature recording. A complete line of standard charts is also available. Brown Instrument Co., Philadelphia, Pa.

## Rusco Slab Stock Brake Lining

Woven of long fiber asbestos and brass wire, the slab is molded to a definite arc with high pressure. The brass wire, being in continuous strands is said to act as a heat conductor in dissipating heat from the drums. High molding pressures are said to result in greater structural strength and greater resistance to water. Slab sizes so designed that many different types of brake shoes can be cut from each slab. Russell Manufacturing Co., Middletown, Conn.

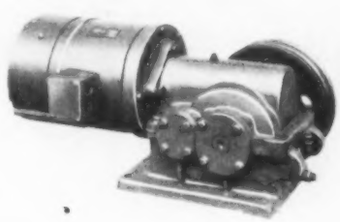
## "Reco" Vibrator

Operating on either d.c. or a.c., can be used as a flasher for lamps up to 100 watts within a range of 200 to 300 flashes per min. Supplied in one size 3x2½x2 in. Reynolds Electric Co., 2650 W. Congress St., Chicago, Ill.

## Seapak

Is a sound and heat insulating material consisting of kapok in sheet form with a thin paper membrane on one side. For insulation of airplanes, trucks, refrigerators. Claimed to be one of the most efficient insulating materials and also one of the lightest of practical insulation materials. Is said to be especially adapted for airplane construction as a temperature and high frequency sound insulation combining lightness in weight and adaptability to construction. Available in two thicknesses, approximately 1/11-in. and ⅛-in. thick. Available also in sheets with a lining of rope paper in various weights. Manufactured in sheets or rolls, 126-in. wide, in lengths up to 150 yd. Cut into sheets if desired. Seaman Paper Co., 410 N. Michigan Ave., Chicago, Ill.

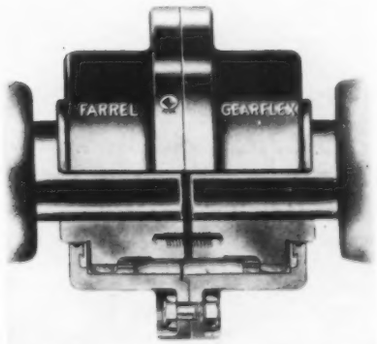
## Star Gear Reducer



With a built-in motor and a combination of worm and spur gear reduction provides any ratio from 25 to 1 to 125 to 1. The motor shaft carries a high angle, high efficiency worm of hardened and ground steel driving a bronze wormwheel. The second reduction is through spur gears. Available with any type of motor, either a.c. or d.c., from ½ to 10 hp. Also available with a built-in magnetically operated disk brake as shown in the illustration. Star Electric Co., Bloomfield, N. J.

## Farrel Gearflex Couplings

A flexible coupling of the gear type consists of two externally geared hubs keyed to the shafts to be connected and engaging an internally geared floating sleeve which encases the two hubs. Misalignment of the two shafts is said to be compensated for by the sleeve assuming a neutral position between the two shafts, eliminating whip or crank action. Accurately generated




external and internal spur gears, cut on the Sykes gear generating machine, are said to insure distribution of the load over a large number of contact surfaces.

Lubricant is carried in a reservoir between the hubs. Rotation of the coupling is said to force oil to the

inner surface of the floating sleeve and to entirely immerse the load-carrying surfaces, providing a cushioning effect to reduce wear. Coupling also available in a single engagement type having one externally geared hub in engagement with an internal gear in the sleeve, the other hub being solid and the sleeve bolted to it. Both types are available in a number of standard sizes and power ratings. Farrel-Birmingham Co., Inc., 344 Vulcan St., Buffalo, N. Y.

## General Electric Phototube

Designated as Type FJ-114 is made in an automobile headlamp bulb and is fitted with the standard single-contact bayonet-type base. A new activation process is said to produce high sensitivity of the usual gas tube in a high vacuum type having an unusual infrared sensitivity. The tube is said to respond to a wave length beyond 12,000 Angstroms corresponding to a temperature of about 500 deg. C. The light-sensitive surface is a caesiated sil-

ver coating of the glas bulb. Electrical contact with sensitive surface is made through platinum foil fused to the inside of the bulb before the coating is applied. General Electric Co., Schenectady, N. Y.

## Goodyear Belts for Multi-V-Drives

Long flexing life, uniform cross-section and low stretch said to result from a construction concentrating the load-carrying capacity about the neutral axis of the belt. One or two endless cords laid in one plane are embedded in rubber. Tension and compression sections of the belt are composed of rubber with layers of fabric distributed through the compression section. The belt is molded to shape, and is inclosed in a fabric envelope so cut that the threads run on the bias, preventing the envelope from taking any part of the load. Close control of pre-stretching during construction is said to insure matching of the belts in each drive and uniform distribution of the load. Worthington Pump & Machinery Corp., Harrison, N. J.

## Fiber Screw Anchors

Consisting of small tubes of jute fiber strands cemented together, said to be unaffected by temperature or moisture. Designed to fit snugly into a hole, the fibers are compressed when the screw is turned in. It is claimed that when properly installed the screw will break out the surrounding material before the plug tears out. Plugs can be used with wood or lag screws for making fastenings in metal without the necessity of tapping. Available for all standard screw sizes from No. 6 wood screws to $\frac{5}{8}$-in. lag screw. It is said that screws can be removed and replaced without damaging the plug, that a smaller hole is required, and that the plug is invisible in the hole. The Rawlplug Co., Inc., 66 W. Broadway, New York, N. Y.

## Arrow Type "CR" Inclosed Switches

For installation on oil burners, refrigerators, small motor appliances, heaters and lighting loads. A positive snap action engages contact blades, providing quick-make and quick-break contacts. Operating lever, which is a part of the mechanism, is made of bakelite and carries floating, self-adjusting phosphor bronze contacts. Said to have ample wiring room and easily accessible connections. Box cover has a lever guard with provisions for locking switch "on" or "off," and box may be sealed. Available for plug or cartridge fuses for use with motors up to 3 hp., in circuits of 30 amp., and 250 volts. Arrow Electric Division, The Arrow-Hart & Hegeman Electric Co., Hartford, Conn.

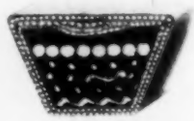
## Westinghouse Photolux Light Control

For turning electric lights on or off without supervision in offices, industrial plants, show windows, schools, signs, street lighting, navigation lights, and airway and airport lighting. Mounted in a cabinet is a vacuum phototube, a one tube amplifier, and a transformer, primary relay, auxiliary contactors, resistors, potentiometers, and condensers. One unit within the cabinet combines the tube sockets and an adjustment panel containing all necessary adjustments. Copper oxide dry rectifiers provide power for the amplifier circuit. Two models are available, a cast aluminum cabinet sealed by gaskets for outdoor application, and the indoor cabinet illustrated. Conduit openings sealed by pipe plugs are provided at the top and bottom of the outdoor model, conduit knockouts are provided on all sides of the case in the indoor model. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

## Rubber-Insulated Cable

Now available in three types for heat-, corona-, or moisture-resistance. For temperatures not exceeding 75 deg. C., Type GE-R359 is recommended for apparatus, transformer lead, motor lead, and power applications for higher temperature service. Said to be suitable for services where a flexible, waterproof cable is required with a non-metallic finish. Moisture-resisting Type GE-R348 is adapted for non-leaded submarine cables. Corona- and moisture-resisting cable Type GE-R351 is recommended for high-voltage transmission, distribution,

Nos 3, 4



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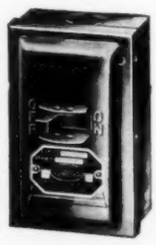


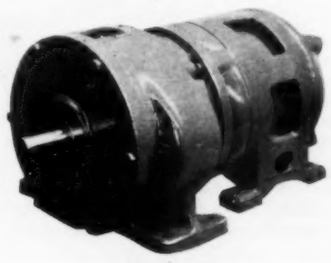
station apparatus, tree wire, submarine, and similar applications. In wet ducts under alternate moisture and dry conditions, where cable is subjected to external heat, in primary networks where leaded cables are not desirable, where braided cables are installed in metallic ducts, where cable is attached to uninsulated clamps are specific applications for this cable. General Electric Co., Schenectady, N. Y.

## Planigressive Speed Reducers

Available in four types, are designed to mount a standard N.E.M.A. motor without change. No alterations of motor shaft are required. All reducers can be supplied with motors or can be mounted on a motor in service. All four types can also be furnished for use with gas engines, water turbines, diesel engines, and can operate in a vertical position with the output shaft up or down. The input and output shafts are concentric in all types. Lubrication is provided by an oil supply in the base; oil seals are used on both shafts.

Type A planetary speed reducers are available in ratios of 3 to 1 up to 7 to 1. For motors up to 3 hp. reducers are mounted on the end bell of the motor, from 5 to 15 hp. gear case has supporting feet. Type B compound speed reducers are available in 12 ratios from 8 to 1 to 24 to 1, designed to mount on standard N.E.M.A. motors. Type C compound speed increasers available in seven ratios from 1.1 : 1 to 5 : 1. Both types designed to mount motors with frames 204 to 326 inclusive. Type D reducer for ratios from 25 to 1 to 600 to 1 is a compound planetary type using composition gears in the first reduction. Reducer cases designed to mount standard flanged motors from ½ to 100 hp. Reducer housing is supplied with feet to take high torques developed with high reduction. Davis & Thompson Co., Milwaukee, Wis.

Type A
Planetary speed reducer

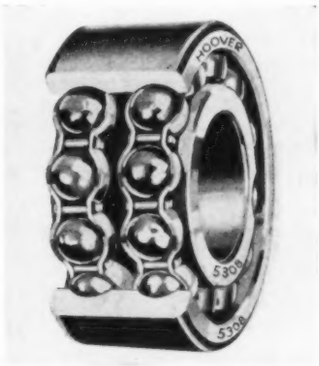


Type B and Type C
Compound speed reducers and increasers

Type D
Compound planetary reducer

## Hoover Double-Row Ball Bearings

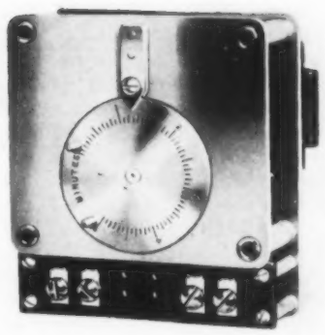
Available in light, medium, and heavy series, made to S.A.E. standards. Bearings are of the deep-groove type without filling slots or interruption of raceways. Said to provide large thrust capacity in either direction in addition to the radial capacity.



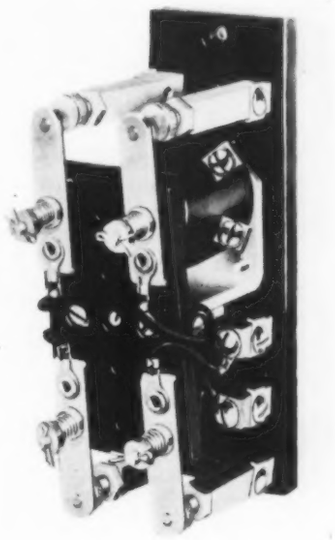

Manufacturing processes are said to insure that each row of balls carries its share of the total load. Formed steel retainers are used for spacing the balls. Electric furnace, high-carbon chrome alloy steel, S.A.E. 2100 is used for all parts of the bearing. Hoover Steel Ball Co., Ann Arbor, Mich.

## Two New Ward-Leonard Relays

A time delay relay designed to produce a definite time interval between two different operations, consists of a small constant speed driving motor, a gear train, a tripping arm and a magnetic clutch between the trip arm and gear train. The time interval is adjusted by changing the arc of travel of the trip arm. Standard gearing provides three time ranges; 4 to 60 sec., 20 sec. to 5 min., 2 min. to 25 min. Input is 5 watts at 110 or 220 volts, 60 cycles. Relay capacity, 2 amp. for a.c. or 0.5 amp. for d.c. Device is approximately 3⅜x3½x3 in.



Time delay relay



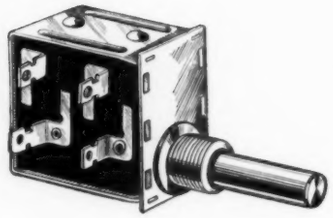
Transfer relay

A transfer relay is available for circuits requiring up and down contacts separate from each other. When the relay operates, one or more circuits are opened and one or more separate and independent circuits are closed. Relay has four poles, two normally open and two normally closed. Stiff metal switch blades or contact fingers, spring shackled and equipped with silver to silver contacts provide sufficient rating for a heavy-duty relay or light-duty contactor service. Ward Leonard Electric Co., Mount Vernon, N. Y.

## Eby Rotary Switches

Multi-polar snap action switches designed to fit into restricted space. Available in either single or double throw with one to six poles. Provided with a steel frame, phosphor bronze,

silver plated double contact and bakelite insulation. Available in types for bushing mounting, as illustrated, or lug mounting. Shafts are plain or with a flat, also provided with a standard knob or toggle lever. The two-pole switch illustrated is 1-in. high, extends 1⅛ in. back of panel, bushing has 3/8-32 thread. Rated at 1 amp., 250 volts; 3 amp., 125 volts. The H. H. Eby Manufacturing Co., Inc., 21st St. and Hunting Park Ave., Philadelphia, Pa.



# NEW BOOKS and PUBLICATIONS

### 1932 Directory of New York State Manufacturers, Volume I

*Issued by the New York State Department of Labor.* Price $3.

This directory covers the 53 counties north of Westchester and Rockland counties, and includes each industry represented by every firm inspected in 1932, with no exclusions on the basis of size. Part I lists approximately 9,000 factories in two registers, one giving all firms in alphabetical order, the other a classified index including more than 100 industry groups, each subdivided by county. The information given for each firm includes office address, location of plants, principal products and, by means of a code letter some indications of size. Also included are summaries and statistical tabulations for each industry, showing the distribution of factories and the number of men and women employed.

Part II includes automobile service and repair, power laundries, dry cleaning and dyeing, each listed with subheadings by county with the same information given for each firm as for those in Part I.

▼

**Balsa Wood**—The Balsa Wood Co., Inc., Brooklyn, N. Y. Booklet, 8 pages, "The Story of Lata Balsa," giving growth and processing, physical characteristics. Applications for sound and heat insulation, buoyancy, available sizes and classifications.

**Bearing Metal**—National Lead Co., 111 Broadway, New York, N. Y. Leaflet, 10 pages, "Satco Bearing Metal," report on the physical properties, temperature characteristics, casting qualities and uses of an improved white metal alloy.

**Cable**—General Electric Co., Schenectady, N. Y. Bulletin GEA-1746, "Glyptal-Cloth Insulated Cable," description and applications.

**Chains** — Link-Belt Co., 300 W. Pershing Road, Chicago, Ill. Souvenir book of the Link-Belt Exhibit at A Century of Progress, 48 pages, "Serving Industry's Needs," with illustrations of more than 150 applications.

**Coupling** — The Falk Corp., Milwaukee, Wis. Bulletin No. 500, 8 pages, "Falk Rawson Four-Duty Coupling," details of construction, operation and typical applications.

**Electric Motors** — Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Circular C-1969, 12 pages, "Westinghouse Thermoguard Motors," with data on thermal protection of electric motors using bi-metal disk thermostat.

**Hydraulic Couplings** — Hydraulic Coupling Corp., 1549 Harper Ave., Detroit, Mich. Booklet, 14 pages, "Hydraulic Couplings," with details on application to variable speed fans and pumps. Description of operation, calculation for determining the size of the coupling, performance curves, capacities and dimension sheets also given.

**Lubricant** — Acheson Oildag Co., Port Huron, Mich. Bulletin No. 102, 4 pages, "The Lubrication of Small Mechanical Devices," typical applications for colloidal-graphited oil.

**Metal Coatings**—Metallizing Co. of America, Inc., 214 Provost St., Jersey City, N. J. Bulletin, 20 pages, "Spray Metal Coatings," describing the process, preliminary treatment, coating characteristics, accessory equipment, and a number of typical applications.

**Needle Bearings**—The Bantam Ball Bearing Co., South Bend, Ind. Booklet, 8 pages, "Engineering Data for Bantam Roller Bearings," typical applications, complete design data, charts and tables.

**Nichrome**—Driver-Harris Co., Harrison, N. J. Data book R-33, 60 pages, "Nichrome," gives complete information on Nichrome V, Comet, Hytemco, Advance, Manganin, radio alloys, nickel steel and nickel. Each alloy described and applications given. Tables of resistance, temperature coefficient, and weight for each material, current-temperature characteristics of straight wire and helical coils, resistance of ribbon, are included. Alignment charts for design of heating units, useful tables of weights and lengths for rounds, squares, flats, hexagons; conversion factors and wire gages, as well as temperature conversion tables, general tables of physical properties of these alloys and pure metals. Price, 25 cents.

**Nickel-clad Steel**—H. Boker & Co., Inc., 101 Duane St., New York, N. Y. Bulletin No. 1, 12 pages, "Methods for Fabrication of Boker Nickel-Clad Steel Sheets, Strips and Wire." Technical and fabrication data on cold working, heating for hot work, drawing, joining, forms, sizes and finishes available, tables of physical properties. Well illustrated information on welding.

**Opacimeter**—Bausch & Lomb Optical Co., Rochester, N. Y. Folder, "To Measure the Opacity of Paper."

**Optical Equipment**—Bausch & Lomb Optical Co., Rochester, N. Y. Catalog, 48 pages, "Lenses, Prisms and Reflectors," foreword on relation between cost and accuracy, details required for ordering. Catalog list includes prices for various types of lenses, prisms and mirrors, and special items.

**Plastics** — Boonton Molding Co., Boonton, N. J. Handbook, 26 pages, "A Ready Reference for Plastics," introduced by a foreword on the particular field of each of the plastics, and a page on production facilities and service available. Mechanical and electrical properties are given for the various types of plastics: molded phenolic, cellulose acetate, ureas, hard rubber, shellac, casein; in molded, cast, or laminated material. Included also are data on molds, color, water absorption, workability, tables of physical properties, and several pages of other useful tables and data.

**Rheostats**—Ward Leonard Electric Co., Mt. Vernon, N. Y. Bulletins Nos. 1103, 1104 and 1105, catalog information on Vitrohm coil-type rheostats, adjustats, and ring-type rheostats, including application, construction, and electrical characteristics.

**Stainless Steels**—United States Steel Corp. Table of physical properties, mechanical properties, and working of seven U.S.S. stainless and heat-resisting steels, compiled by the Research Laboratories of the U.S. Steel Corp.

**Steel** — Latrobe Electric Steel Co., Latrobe, Pa. Handbook and catalog, 204 pages, listing applications, heat-treatment, forging, grinding, classification for high-speed and carbon steel, die, magnet, chrome vanadium and stainless steel; also, standard tables of weights, gages, equivalents, melting points.

**Steel**—Joseph T. Ryerson & Son, Inc., Chicago, Ill. Stock list, 230 pages with information and data on standard and special grades of cold finished steels, alloy steels, stainless steels, and brass, copper and other non-ferrous metals.

# Safe Loads for Needle Bearings

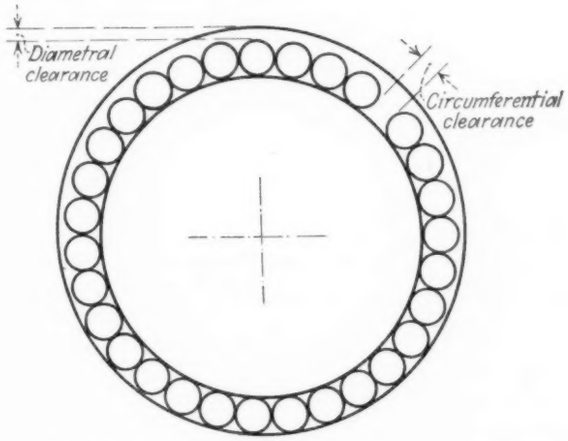
*Contributed by Norman M. Wickstrand*

The accompanying alignment chart provides an accurate, convenient method for determining the steady loads that can be carried safely by needle bearings of a given size, or for determining the diameter, length and number of needles when the other factors are known or assumed. The chart applies to needles of either domestic or foreign manufacture.

For shafts 1 in. in diameter, needle diameter should be $\frac{3}{32}$ in. or larger and proportionately larger needles are required for larger shafts. Under conditions of only partial rotation such as wrist-pin bearings, the needle diameter should be small enough to provide at least one full rotation of the needle to minimize localized wear on either the needles or the raceways.

Needles longer than 3 to 10 times the diameter have a tendency to get out of line. When a greater length is needed, two or more sets of rollers should be used, each set separated by hardened and ground retaining rings or shoulders. When two or more sets of needles are used together, the capacity of the bearings is determined from their combined length or from the total of each set. Hardened and ground surfaces should be provided at each end of each set of needles.

Longitudinal clearances of from 0.010 to 0.030 in. are required for each set of rollers. The diameter and number of needles is chosen to provide a circumferential clearance of at least 0.0001 in. per roller up to a maximum of nearly the diameter of one roller for the entire set. Clearances are figured on the high limit for the needles.




**The definitions of diametral clearance and circumferential clearance when referring to the design of needle bearing mountings are as indicated in this figure**

LIMITS OF DIAMETRAL CLEARANCE FOR NEEDLE BEARING ASSEMBLY

| Shaft Diameter | Diametral Clearance | |
|---|---|---|
| | Minimum | Maximum |
| Up to 2 in. | 0.0005 in. | 0.0015 in. |
| 2 to 4 in. | 0.0010 in. | 0.0020 in. |
| 4 to 6 in. | 0.0015 in. | 0.0025 in. |

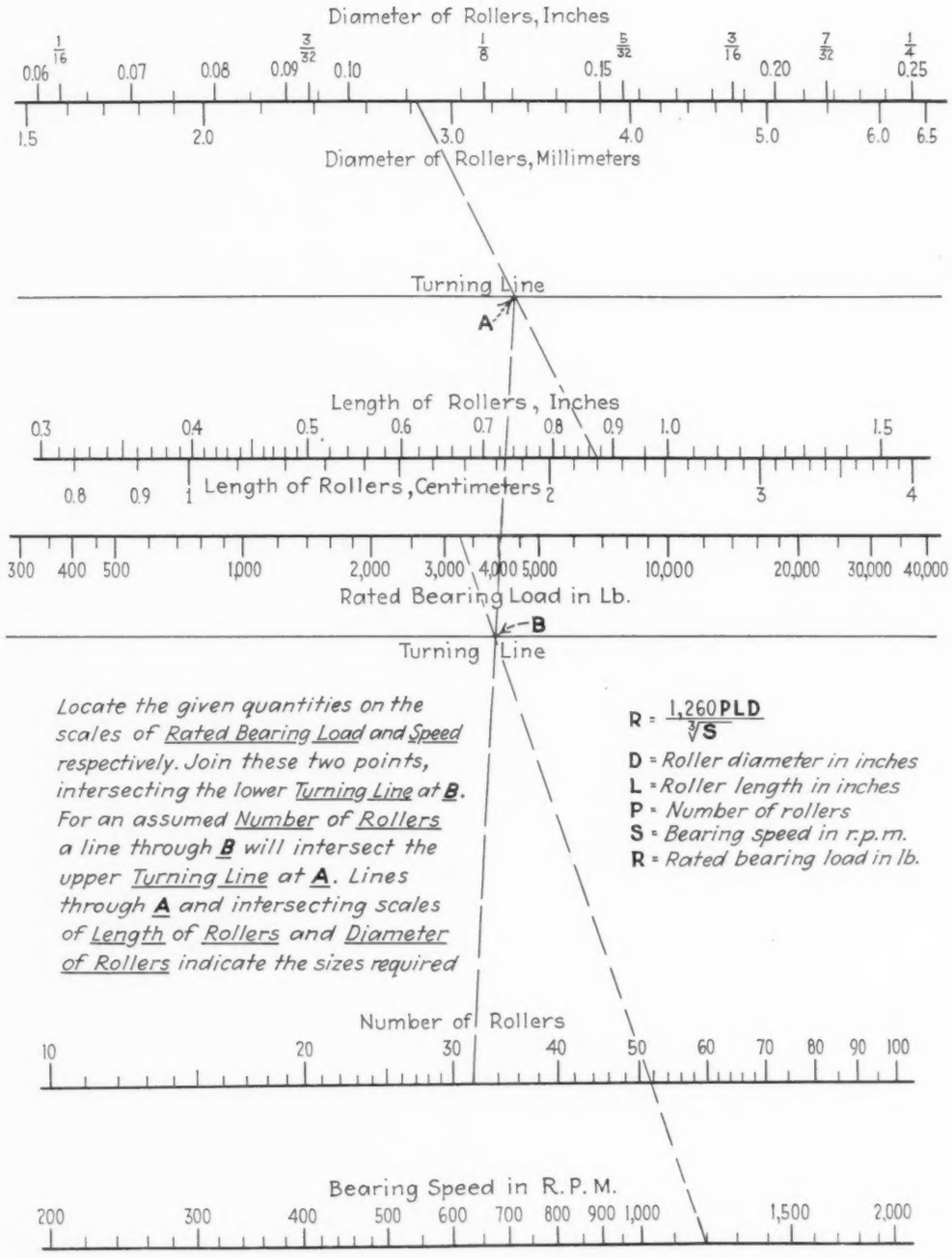
Shaft diameter tolerances are within plus nothing to minus 0.0005 in. Tolerances for the outer race are held within plus 0.0005 to minus nothing. The diameter of the outer race is the shaft diameter plus twice the needle diameter plus the diametral clearance given in the table. Diametral clearances are based on the high limit for diameter of shaft and needles and low limit for bore of outer race. If the minimum diametral clearance is reduced there is danger of preloading that will seriously affect the life and capacity of the bearing.

In order to obtain full capacity both the inner race or shaft and the outer race must be hardened and ground. Carburizing to a uniform depth provides a soft and ductile core, an advantage under some conditions. To produce a case of uniform depth free from soft spots, oil quenching is recommended. Raceways should have a minimum hardness of 60 to 63 Rockwell "C" scale.

Any lubricant suitable for ball bearings or cage-type roller bearings is suitable for needle bearings. As the principal function of the lubricant is to preserve the surface of the needles and races from rust or corrosion, lubricants should be free from acid, alkali, sulphur or other corrosive materials. Needle bearings designed for use with a heavy lubricant require the maximum values for diametral and circumferential clearances.

Available needle sizes in both millimeter and fractional-inch sizes can be obtained from manufacturers. For typical installations of needle bearings refer to "Applications of Small Rollers for Bearings," by K. L. Herrmann, *Product Engineering*, April, 1933, p. 138.

# Safe Loads for Needle Bearings

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